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AIR AND GAS COMPRESSION

BY

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PREFACE

Recent researches in the properties of gases, especially in regard to compressibility, critical data, and specific heats, have prompted me to attempt to apply some of the results attained to the solution of the problems of air and gas compression. Although I realize that there is still much to be learned about the behavior of gases, especially under high pressure, I feel that I have gathered together sufficient data to warrant the publication of a new textbook on the theory of compression, and hope that it will prove useful in stimulating further thought along the lines which I have laid out.

The application of compressibility correction factors to the fundamental compression equations is of recent origin, perhaps growing out of similar developments in the measurement of gas by orifice meter. It is evident that the theories of compression and of gas measurement should go hand in hand, and that no refinements in the knowledge of compression can be made without adequate and accurate metering facilities.

In addition to the theory of compression, this book also contains a chapter on the flow of gas in pipe lines, which is based on the Weymouth equation. A new series of tables is included which greatly reduces the labor required in computing pressure drop in pipe lines. The chapter on the orifice meter serves merely as an introduction to the subject, containing only enough information to enable the reader to use the gas-measurement handbooks published by the meter manufacturers and technical societies.

The solution of problems is based on equations, tables, and alignment charts — an arrangement which will enable the reader to check nearly all computations in several different ways, and which, it is hoped, will greatly increase the usefulness of this volume.

Thanks are due to Mr. Lyman F. Scheel, of the Union Oil Co. of California, and to Mr. A. K. Hegeman, of Smith-Booth-Usher Co., Los Angeles, for valuable data and suggestion.

THOMAS T. GILL

SANTA FE SPRINGS, CALIFORNIA
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CONTENTS

TABLES

ALIGNMENT CHARTS

AIR AND GAS COMPRESSION

CHAPTER I

DEFINITIONS AND FUNDAMENTAL UNITS

The ordinary bellows, used by blacksmiths and by the early smelters of iron and other metals, was perhaps the first form of the air compressor. The intake valve usually consisted of several holes in the wooden frame, over which were placed flaps of cloth or leather. Some sort of rudimentary check valve was provided on the discharge, to prevent air from being taken in from the wrong direction on the suction stroke.

The plunger pump, originally designed as a *flamenwerfer*, or flame-thrower, for a weapon in the Thirty Years' war, is credited[1] to Otto von Guericke (1602–1686). Although used for pumping oil which was ignited on leaving a nozzle, von Guericke's apparatus had all the essential parts of a compressor. In fact, it was later employed as an air pump, to evacuate the air from two large hollow hemispheres placed face to face. It is related that eight horses were unable to pull the two hemispheres apart, until the vacuum was released.

Most of the early researchers in the science of pneumatics were concerned with the vacuum. The crushing, by atmospheric pressure, of flimsy vessels from which the air had been partially evacuated, gradually aided in the realization that air has weight — a fact which was known to Plato and Aristotle.[2] Galileo (1564–1642) estimated that the ratio of the weight of air to that of water was about 1 to 400. Torricelli, a pupil of Galileo, demonstrated that the atmosphere will support a column of mercury approximately 30 in. in height, thus inventing the first barometer. See Fig. 1. In 1647, Pascal repeated Torricelli's experiment, substituting a column of water 34 ft. high for the mercury.

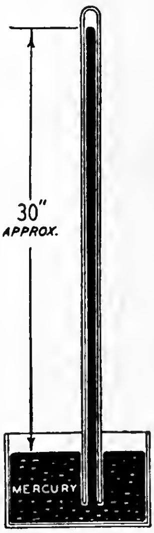


FIG. 1. Barometer.

[1] *Development of Physical Thought*, by Loeb and Adams, p. 104, John Wiley & Sons, 1933.

[2] *History of Physics*, by Florian Cajori, p. 80, Macmillan Co., 1929.

Units. In order to conform to usage in the industry, all calculations in this book will be based on the English system of units, as they are called, which includes the foot, the pound avoirdupois, and the Fahrenheit temperature scale. We will be concerned principally with the double-acting, reciprocating type of compressor, although the fundamental theoretical equations for this type of machine, within certain limits, may also be applied to centrifugal compressors and blowers.

Gases. As distinguished from solids and liquids, a gas represents a state of matter in which the molecules are in rapid motion with little regard or attraction for each other.[1] Gases do not have any definite form or shape. When a gas is compressed, the molecules are crowded closer together. The reverse is true when a gas expands.

Temperature. The velocity of the molecules of a gas varies with the temperature, the molecules moving faster when the gas is hot than when it is cold. The kinetic energy of translation is a measure of the relative hotness or coldness of a gas, and therefore of the property known as temperature.

On the Fahrenheit scale, under standard atmospheric pressure, water boils at 212° and ice melts at 32°. These two points correspond to 100° and 0° Centigrade. The temperature range between boiling water and melting ice, on the Fahrenheit scale, is 180° at 760 mm. of mercury barometer. A degree Fahrenheit is thus 1/180 of the difference in temperature between these two basic points.

Absolute Temperature. It has been observed that the so-called perfect gases shrink upon cooling, when subjected to a constant pressure. The amount of this shrinkage is proportional to the number of degrees above a hypothetical point known as absolute zero, which is given in textbooks as −459.72°F., or 459.72° below zero on the Fahrenheit scale. This corresponds to −273.18° Centigrade.

Theoretically, all gases would have zero volume at absolute zero, but, as a matter of fact, they all liquefy before this temperature is reached. The kinetic theory holds, however, that, at absolute zero, the molecules of a gas have no kinetic energy, as all molecular movement is presumed to have ceased.

Temperatures used in gas calculations are measured from absolute zero, which for convenience will be taken as −460°F. To obtain absolute temperatures, it is necessary to add 460° to the values obtained from the Fahrenheit thermometer.

The standard base or reference temperature in compression work and gas measurement will be taken as 60°F. (520°F. abs.)

[1] For a complete treatment of kinetic theory, see *Kinetic Theory of Gases* by Kennard, McGraw-Hill Book Co., 1938.

Pressure. Pressure is force applied to, or distributed evenly over, a surface, and is expressed in magnitude as force per unit area. Molecular theory ascribes the cause of gas pressure to the striking energy of the molecules in any given space or unit of volume.

The ordinary Bourdon or spring type of pressure gauge records pressure above atmosphere and is usually calibrated in pounds per square inch. All gauge pressures in this book will be so given unless otherwise indicated. Any pressure below that of the local atmosphere is regarded as a vacuum and is ordinarily measured in inches of mercury by a U-tube or manometer, one end of which is open to the air. Special combination spring pressure gauges are also used, which are calibrated in pounds per square inch above atmosphere and in inches of mercury vacuum.

Absolute Pressure. For calculation purposes, it is customary to begin the measurement of pressure at absolute zero, which is the theoretically perfect vacuum. The absolute pressure at any point is the algebraic sum of the pressure caused by the atmosphere and the additional pressure shown on the gauge or manometer. If the atmospheric pressure given by the barometer is 30 in. of mercury, or 14.73 lb. per sq. in., and the pressure gauge reads 9 lb. per sq. in., the absolute pressure is $14.73 + 9 = 23.73$ lb. per sq. in. If the manometer or vacuum gauge reads 3 in. of mercury vacuum, the absolute pressure is $14.73 - (3 \times 0.491) = 13.26$ lb. per sq. in. Refer to Fig. 2.

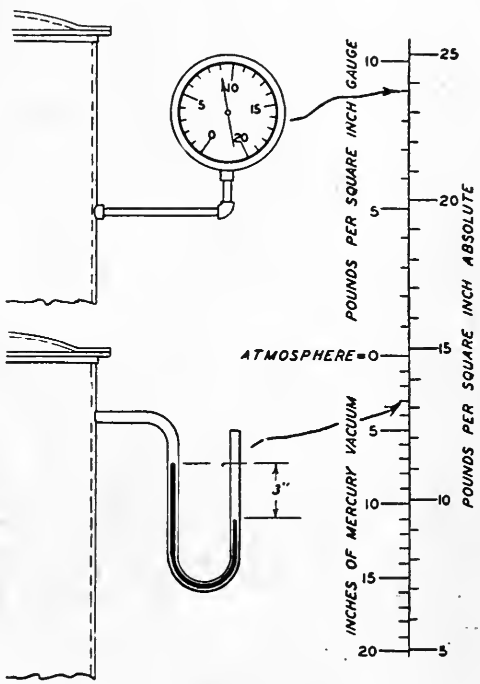


FIG. 2.

NOTE: To convert inches of mercury to pounds per square inch, multiply by 0.491.

Atmospheric Pressure. The normal atmospheric pressure at sea level is usually given as 14.696 lb. per sq. in., or 2116.2 lb. per sq. ft.[1] This is equal to the pressure exerted by a column of mercury

[1] *Handbook of Chemistry and Physics* by Hodgman, 23rd ed., p. 1933, Chemical Rubber Publishing Co., 1939.

760 mm. high or, in English units, to a barometer of 29.92 in. In both cases, the height of the mercury column is corrected to 0°C. or 32°F. In European practice, pressures are usually expressed in terms of this standard atmosphere.

Since July 1, 1939, the U. S. Weather Bureau[1] has reported barometric pressures in a unit known as the millibar, which is 1/1000 of the metric unit, the bar, which in turn is defined as 1,000,000 dynes per square centimeter, or 14.504 lb. per sq. in. abs. Thus 1015.9 millibars equal 30 in. of mercury, or 1 millibar equals 0.0295 in. of mercury.

The standard atmospheric pressure in compression and gas measurement will be taken as 14.73 lb. per sq. in. abs., 30 in. of mercury, or 1016 millibars. This standard has been adopted by the American Gas Association, the Pacific Coast Gas Association, the California Natural Gasoline Association, and others. The standard of 14.7 lb. per sq. in. often seen in textbooks of thermodynamics is much used for air measurement. In the natural-gas industry, a unit sometimes met with assumes an atmospheric pressure of 14.4 lb. per sq. in. and allows 0.25 lb. pressure drop to actuate the meter, making a total pressure of 14.65 lb. per sq. in. abs.

Volume. The cubic foot is the fundamental unit of volume for gases. The standard cubic foot will be measured at 60°F. (520°F. abs.) and 14.73 lb. per sq. in. abs. (30 in. of mercury barometer.) Compressor capacities will be given in standard cubic feet per minute, and also in thousands of standard cubic feet per 24 hours (abbreviated M.c.f.). The former unit is common in air-compression work, and the latter in the natural-gas industry.

Density. Mass per unit volume is the universal definition of density. This may be expressed in any desired units, such as pounds per cubic foot. For convenience, if densities are referred to certain well-known substances as standards, the density relation is known as the specific gravity.

Specific Gravity. In this book, specific gravities of gases will be referred to air, both substances being weighed and measured under identical conditions. The specific gravity of air will thus become 1.00. Specific gravity, being merely a ratio, is naturally without units of any kind, and therefore is spoken of as being dimensionless.

On account of the deviation from the ideal gas laws, the specific gravity of gases is not constant for all pressures. In practice, specific gravities are usually measured at the standard conditions 30 in. of mercury and 60°F.

[1] "The Use of Millibars," Form 4090, *Misc. Publ.*, U. S. Dept. Agr., Weather Bureau.

Pound Mol. A pound mol of any substance is the molecular weight of the substance expressed in pounds. Thus, a pound mol of oxygen, with a molecular weight of 32, weighs 32 lb.

Heat. The British thermal unit (B.t.u.) is usually taken as the amount of heat required to raise the temperature of 1 lb. of water through 1°F. More exactly, it is 1/180 of the heat required to raise the temperature of 1 lb. of water at atmospheric pressure from 32°F. to 212°F. without change of state.

Specific Heat. The specific heat of a substance is the dimensionless ratio of its thermal capacity to that of water. Since, for all practical purposes, the thermal capacity of water is unity, or 1 B.t.u. per lb. per °F., we may take the specific heat of any other substance as the amount of heat required to raise the temperature of 1 lb. of that substance through 1°F.

Specific heats of nearly all substances vary widely with temperature, so that, when a specific heat is given, the temperature range through which it applies must also be specified.

When solids and liquids undergo a change in temperature, the specific heat of the substance in question depends entirely upon the change of molecular energy that takes place. In gases, however, the process is much more complicated,[1] as external work of expansion may be done at the expense of some of the internal energy. As it is impossible to describe every variety of temperature change that gases undergo, it is convenient to say that gases have two specific heats: C_p at constant pressure, and C_v at constant volume. Of these, C_p is always the greater, because a larger amount of energy or heat must be supplied for a given temperature change to replace the energy used up in the external work of expansion. C_v for a similar temperature change would be smaller than C_p by the amount of external work involved, as it is obvious that no external work or expansion can occur at constant volume.

The ratio C_p/C_v, which is generally known as k or γ, occurs in equations for the adiabatic compression or expansion of gases. The value of this ratio is not constant but is a function of critical pressures and temperatures.[2]

Ratio of Compression. The relation between the final and initial pressures during compression, expressed in absolute units, is known as the ratio of compression.

[1] For a more complete discussion, see *Handbook of Engineering Fundamentals* by Eshbach, p. 7–16, John Wiley & Sons, 1936; and *Fluid Mechanics* by Dodge and Thompson, p. 397, McGraw-Hill Book Co., 1937.

[2] See "Specific Heat Ratios for Hydrocarbons" by W. C. Edmister, *Ind. Eng. Chem.*, vol. 32, No. 3, p. 373, March, 1940.

Isothermal Changes. When a gas undergoes a change of pressure or volume at constant temperature, the change is said to be isothermal.

Adiabatic Changes. A gas is said to undergo an adiabatic change when its condition is altered without gain or loss of heat, and thus at constant entropy.

Critical Temperature. The temperature above which a pure gas cannot be liquefied by pressure alone is known as the critical temperature.

Critical Pressure. The pressure under which a pure gas may exist in a gaseous state in equilibrium with its liquid at the critical temperature is known as the critical pressure. The definitions of critical temperature and pressure apply to pure or homogeneous gases only.

Critical Volume. The specific volume of a pure gas at the critical temperature and the critical pressure, which may be stated in cubic feet per pound, is known as the critical volume.

Common Gases. Properties of some of the more important gases met with in compression practice are given in Table I.[1]

Air is a mixture of oxygen and nitrogen, with small quantities of carbon dioxide, water vapor, hydrogen, and the rare gases argon, neon, helium, etc. Its composition varies with locality, and also with altitude above sea level. Under normal conditions, an average analysis would show: nitrogen, 78 per cent; oxygen, 21 per cent; argon, 0.9 per cent; carbon dioxide, 0.03 per cent; hydrogen, 0.01 per cent; rare gases, 0.06 per cent.

The principal constituents of natural gas are members of the paraffin hydrocarbon series, which have the type formula $C_nH_{(2n+2)}$. Methane, with a boiling point of -258°F. at atmospheric pressure, and ethane (boiling point -127°F. at atmospheric pressure), are met with as gases under ordinary conditions, and make up the bulk of natural gases.

Compression. The problem of compression is essentially one of the flow of fluids, as the gases being compressed must be transported through the compressor, from the intake to the discharge lines. The change in velocity energy between the stream of gas entering at the intake and leaving at the exhaust is proportional to the squares of the velocities at the points in question. This change in energy, being very small in comparison with the work of compression, will ordinarily be neglected in calculations.

[1] Specific heat and critical data mostly from *International Critical Tables*, vol. 3, pp. 248–249, and vol. 5, pp. 80–82, McGraw-Hill Book Co., 1926.

For accurate values of C_p/C_v to three decimal places, and a thorough analysis of experimental methods employed in their determination, see *Specific Heats of Gases* by Partington and Shilling, E. Benn, Ltd., London, 1924.

See also "Physical Constants of the Components of Natural Gas and Gasoline," *Bulletin* TS-401, California Natural Gasoline Association, Los Angeles, 1940. This bulletin contains complete tables and an exhaustive bibliography.

For all compression calculations in this book no account is taken of condensation or partial liquefaction of the gases during the compression processes. For a consideration of this subject, see "Compression of Refinery and Casinghead Gases" by W. J. Murray, *Ind. Eng. Chem.*, October, 1924.

TABLE I

PROPERTIES OF GASES

Gas	Formula	Molecular Weight	Boiling Point at 1 Atmosphere, °F.	Specific Gravity [Air = 1.00]	Ratio C_p/C_v at 1 Atmosphere and 15°C.	Critical Temperature, °F.	Critical Pressure, lb. per sq. in. absolute
Air		28.97		1.00	1.40(17°C.)	−221	546
Methane	CH_4	16.03	−259	0.553	1.31	−116	673
Ethane	C_2H_6	30.05	−127	1.037	1.22	90	717
Propane	C_3H_8	44.06	− 44	1.521	1.14	204	632
Isobutane	C_4H_{10}	58.08	10	2.005		273	544
N-butane	C_4H_{10}	58.08	31	2.005	1.11	307	529
Isopentane	C_5H_{12}	72.09	82	2.488		370	482
N-pentane	C_5H_{12}	72.09	97	2.488	1.09	387	485
N-hexane	C_6H_{14}	86.11	156	2.972	1.08	455	434
N-heptane	C_7H_{16}	100.12	209	3.456	1.07	512	394
N-octane	C_8H_{18}	114.14	259	3.940	1.06	565	362
Helium	He	4.00	−452	0.137	1.66(−180°C.)	−450	33
Oxygen	O_2	32.00	−297	1.105	1.40	−182	730
Nitrogen	N_2	28.02	−320	0.970	1.40	−233	492
Hydrogen	H_2	2.016	−423	0.0696	1.41	−400	188
Carbon dioxide	CO_2	44.00	−109	1.520	1.30	88	1073
Sulphur dioxide	SO_2	64.065	14	2.213	1.29	315	1142
Ammonia	NH_3	17.031	− 28	0.590	1.31	270	1638
Methyl chloride	CH_3Cl	50.481	− 11	1.744	1.20	289	967
Ethyl chloride	C_2H_5Cl	64.51	56		1.13	369	764

GENERAL BIBLIOGRAPHY ON COMPRESSION

Frank Richards, *Compressed Air Practice,* McGraw-Hill Book Co., 1913.

Elmo G. Harris, *Compressed Air,* McGraw-Hill Book Co., 1917.

Simmons, *Compressed Air,* McGraw-Hill Book Co., 1921.

Eshbach, *Handbook of Engineering Fundamentals,* John Wiley & Sons, 1936.

Gas Engineers' Handbook, McGraw-Hill Book Co., 1934.

Lucke and Flather, *Engineering Thermodynamics,* McGraw-Hill Book Co., 1915.

Lester C. Lichty, *Natural Gas,* John Wiley & Sons, 1924.

Sage and Lacey, " Compression of Hydrocarbon Gases," Paper presented before the California Natural Gasoline Association at April meeting, 1939, and published in *California Oil World and Petroleum Industry,* second April issue, 1939.

CHAPTER II

BAROMETRIC PRESSURE AND ALTITUDE ABOVE SEA LEVEL

Air compressors are vitally affected by changes in atmospheric pressure. These changes may reach 3 per cent above or 5 per cent below the normal, but such extremes are rarely of long duration.[1]

During special tests, barometric readings are desirable, but for ordinary compressor operation, it is accurate enough to adopt some method of obtaining a standard pressure corresponding to the altitude of the compressor in question, as the labor of taking daily and hourly barometer readings would be out of the question. The following formula[2] may serve as a basis for obtaining standard atmospheric pressures as a function of elevation above sea level:

$$Z = \left[1 + \left(\frac{t_1 + t_2 - 64}{900}\right)\right] \times 52{,}494 \left(\frac{B_1 - B_2}{B_1 + B_2}\right) \qquad (1)$$

where Z = difference in height of two stations in feet.
B_1 = barometric pressure of lower station in inches of mercury.
B_2 = barometric pressure of upper station in inches of mercury.
t_1 = temperature of lower station in degrees Fahrenheit.
t_2 = temperature of upper station in degrees Fahrenheit.

Assuming the temperature of the upper and lower stations to be 60°F., and taking the barometer at sea level to be 30 in. of mercury, the above equation reduces to

$$Z = \left(\frac{14.73 - B}{14.73 + B}\right) \times 55{,}760 \qquad (2)$$

where B is given in pounds per square inch. With this equation, it is possible to compute the altitude above sea level corresponding to any atmospheric pressure, and thus the standard pressure for any given altitude.

Table II gives factors for various altitudes and temperatures, based on ordinary pressure and temperature relations, and also on equation 2. Tabular values represent the number of standard cubic feet (14.73 lb.

[1] For barometric variations at New York, see *World Almanac* for 1941, p. 186, New York World-Telegram.

[2] *Smithsonian Physical Tables*, 7th Ed., p. 145.

per sq. in. and 60°F.) contained in 1 cu. ft. at the conditions specified. The pressures or barometers given in this table are computed for the centers of the corresponding zones. The average temperature is taken as 60°F. for both upper and lower stations. If the yearly mean temperature varies more than 10° from this figure, it might be desirable to

TABLE II

ALTITUDE–TEMPERATURE FACTORS FOR GAS VOLUMES

Altitude in Feet above Sea Level		Atmospheric Pressure		Temperature, °F.									
		Lb. per sq. in.	In. of mercury	0°	20°	40°	60°	70°	80°	90°	100°	120°	140°
0	0	14.73	30.00	1.130	1.083	1.040	1.000	0.981	0.963	0.945	0.928	0.896	0.866
0	150	14.7	29.93	1.128	1.081	1.038	0.998	0.979	0.961	0.943	0.926	0.894	0.865
150	340	14.6	29.73	1.120	1.073	1.031	0.991	0.972	0.954	0.937	0.920	0.888	0.859
340	535	14.5	29.53	1.113	1.066	1.023	0.984	0.966	0.948	0.930	0.914	0.882	0.853
535	730	14.4	29.32	1.105	1.059	1.016	0.977	0.959	0.941	0.924	0.907	0.876	0.847
730	920	14.3	29.12	1.097	1.051	1.009	0.971	0.952	0.935	0.918	0.901	0.870	0.841
920	1,120	14.2	28.91	1.089	1.044	1.002	0.964	0.946	0.928	0.911	0.895	0.864	0.835
1,120	1,320	14.1	28.71	1.082	1.037	0.995	0.957	0.939	0.922	0.905	0.889	0.858	0.829
1,320	1,510	14.0	28.51	1.074	1.029	0.988	0.950	0.932	0.915	0.898	0.882	0.852	0.823
1,510	1,710	13.9	28.30	1.067	1.022	0.981	0.943	0.926	0.908	0.892	0.876	0.846	0.817
1,710	1,920	13.8	28.10	1.059	1.015	0.974	0.937	0.919	0.902	0.886	0.870	0.840	0.812
1,920	2,120	13.7	27.90	1.051	1.008	0.967	0.930	0.913	0.895	0.879	0.863	0.834	0.806
2,120	2,330	13.6	27.69	1.043	1.000	0.960	0.923	0.906	0.889	0.873	0.857	0.828	0.800
2,330	2,535	13.5	27.49	1.036	0.993	0.953	0.916	0.899	0.882	0.866	0.850	0.821	0.794
2,535	2,740	13.4	27.28	1.028	0.985	0.946	0.909	0.892	0.876	0.860	0.844	0.815	0.788
2,740	2,950	13.3	27.08	1.020	0.978	0.939	0.902	0.885	0.869	0.853	0.838	0.809	0.782
2,950	3,160	13.2	26.88	1.013	0.971	0.932	0.896	0.879	0.863	0.847	0.832	0.803	0.776
3,160	3,370	13.1	26.67	1.005	0.963	0.925	0.889	0.872	0.856	0.840	0.825	0.797	0.770
3,370	3,590	13.0	26.47	0.997	0.956	0.918	0.882	0.866	0.850	0.834	0.819	0.791	0.765
4,000	5,100	12.5	25.45	0.959	0.919	0.882	0.848	0.832	0.817	0.802	0.788	0.761	0.735
5,100	6,300	12.0	24.43	0.921	0.882	0.847	0.814	0.799	0.784	0.770	0.756	0.730	0.706
6,300	7,500	11.5	23.42	0.882	0.846	0.812	0.781	0.766	0.752	0.738	0.725	0.700	0.676
7,500	8,700	11.0	22.40	0.844	0.809	0.776	0.747	0.732	0.719	0.706	0.693	0.669	0.647
8,700	10,000	10.5	21.38	0.806	0.772	0.741	0.713	0.699	0.686	0.674	0.662	0.639	0.618
10,000	11,300	10.0	20.36	0.767	0.735	0.706	0.679	0.666	0.654	0.642	0.630	0.608	0.588

$$\text{Factor} = \frac{520}{14.73} \times \frac{P}{T}$$

where P = absolute pressure of gas in pounds per square inch.

T = absolute temperature of gas in degrees Fahrenheit.

compute a more accurate value by using equation 2. Once decided upon, a value for atmospheric pressure may be taken for any given compressor plant and used for all calculations throughout the year, unless seasonal variations in temperature are excessive.

Table II also gives a comparison of the intake efficiency of air compressors at different altitudes. Thus a machine operating at 40°F. and 10 lb. per sq. in. abs. at approximately 11,000 ft. above sea level would deliver only 70.6 per cent as much air as the same machine at sea level and 60°F., if operated at the same speed under both sets of conditions.

Comparison of the absolute pressure of the atmosphere, and the effect of altitude on pressure measurement, are given in Fig. 3 for Los Angeles, 270 ft. above sea level, with an average atmospheric pressure of 14.6 lb. per sq. in. abs.; Cheyenne, Wyoming, altitude 6062 ft., 11.8 lb. per sq. in.; and the summit of Pikes Peak, elevation 14,108 and pressure 8.8 lb. per sq. in.

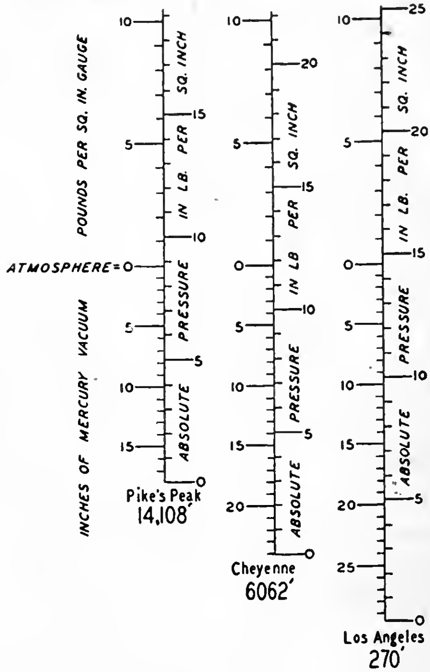


FIG. 3.

NOTE. This chapter is based on a paper by B. M. Laulhere and W. M. Young, "Recommendation for Standardization of Barometric Pressure in Natural Gas Measurement," presented at the November, 1927, meeting of the California Natural Gasoline Association, Los Angeles, California.

CHAPTER III

FUNDAMENTAL GAS LAWS

The principal variables of any gas are its temperature, pressure, and volume. Changes in any of these may induce changes in some or all of the others.

Boyle's law, credited to Robert Boyle (1627–1691), states that the volume occupied by a given weight of any gas at constant temperature varies inversely as the absolute pressure to which it is subjected. This law is sometimes credited to a Frenchman, Edme Mariotte (1620–1684), who performed his experiments subsequently to those of Boyle, but who published his results in his treatise, *Sur la nature de l'air,* in 1676,[1] before Boyle's work became known.

As an illustration of this law, let us consider that a certain weight of gas occupies 6 cu. ft. at a pressure of 15 lb. per sq. in. abs. At 30 lb. abs. the same gas would occupy 6 (15/30) = 3 cu. ft., provided that there had been no change in temperature.

The law of Charles,[2] or Gay-Lussac,[3] states that the volume of a given weight of any gas at constant pressure varies in direct proportion to its absolute temperature. Thus, if a given weight of gas occupies 1 cu. ft. at 0°F. (460°F. abs.), at 115° (575°F. abs.) the same gas will occupy 575/460 = 1.25 cu. ft., provided that there has been no change in pressure.

Boyle's law may be expressed mathematically as

$$P_1V_1 = P_2V_2 = P_3V_3 = K \quad \text{(a constant)} \tag{3}$$

where the subscripts refer to different conditions of the same weight of gas, and the pressures are taken in absolute units.

Adding the effect of temperature,

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} = \frac{P_3V_3}{T_3} = K \quad \text{(a constant)} \tag{4}$$

where T is the absolute temperature of the gas in degrees Fahrenheit.

[1] *History of Physics*, by Florian Cajori, p. 79, Macmillan Co., 1929.

[2] Jacques Charles, 1746–1823.

[3] Louis-Joseph Gay-Lussac, 1778–1850.

Equations 3 and 4 refer to different conditions of the same weight of gas. To introduce a term for the weight of gas, we may write equation 4 without subscripts as $PV/T = K$. The constant K must contain some term for the weight and density of the gas, as well as a dimensional constant, depending on the units employed. Most textbooks on thermodynamics express the relationship[1] as

$$PV = wRT \tag{5}$$

where w is the weight of the gas, and R is a constant for the gas being considered. Equation 5 expresses what is known as the "perfect gas law."

In textbooks on chemical engineering,[2] we find equation 5 expressed as $PV = 1544\, n\, T$, where P is the absolute pressure of the gas in pounds per cubic foot; V is in cubic feet: and n is the number of pound mols of the gas in question.

Neither of these two equations can be used directly unless the molecular weight or composition of the gas is known. They are not convenient to use in gas mixtures, such as we encounter in natural-gas practice.

It is desired to reduce the last equation quoted above to a form using a term for the specific gravity of the gas, based on air as a standard.

If w is the weight in pounds of the gas being considered, then

$$n = \frac{w}{28.97\, G}$$

where 28.97 is the molecular weight of air and G is the specific gravity (air = 1.00).

Substituting in the original equation, and dividing by 144 to reduce to pounds per square inch,[1]

$$PV = \frac{1544\, w\, T}{144\, G\, 28.97}$$

$$PV = \frac{0.37\, w\, T}{G} \tag{6}$$

where P is the pressure of the gas in pounds per square inch; V is the volume in cubic feet; w is the weight in pounds; T is the absolute temperature of the gas in degrees Fahrenheit: and G is the specific gravity, based on air = 1.00.

[1] See Chapter XIII.

[2] *Principles of Chemical Engineering* by Walker, Lewis, and McAdams, p. 6, McGraw-Hill Book Co., 1923.

Equation 6 may be used for problems relating to any weight of gas, provided that the specific gravity is known.

Table III gives pressure factors for gas volumes, based on a local atmospheric pressure of 14.6 lb. per sq. in. abs., which corresponds to an altitude of about 250 ft. above sea level. The table gives the number of standard cubic feet, measured at 14.73 lb. per sq. in. abs., contained in 1 cu. ft. at the given pressure. To use the table, multiply the pressure factor by the number of cubic feet at the given pressure, and obtain standard cubic feet. Thus, 55 cu. ft. of air at 26 lb. per sq. in. gauge will equal $55 \times 2.76 = 151.58$ standard cubic feet. The table also gives the approximate "ratio of compression" for air compressors operating near sea level. The ratio of compression is obtained by dividing the final by the initial pressure, both in absolute units. Table III takes no account of deviation from the ideal gas laws, which will be considered in Chapter XIII.

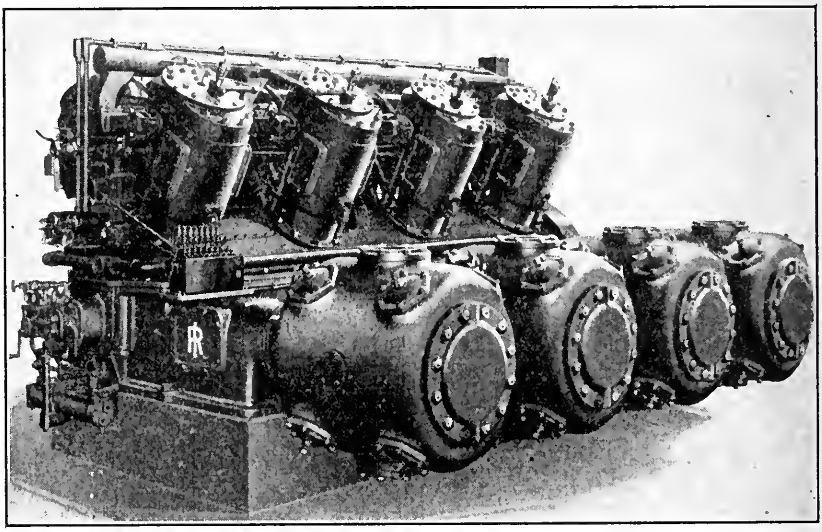


FIG. 4. Ingersoll-Rand 8-XVG 300-hp. four-cycle V-type gas-engine-driven compressor.

In actual compression or expansion, the pressure-volume relationship, $PV = K$ (equation 3), holds good only if the change takes place at constant temperature. If the temperature is allowed to change, the volume of the gas will be affected. Nearly all compressors are water cooled to keep the process of compression at as low a temperature as possible.

The change of temperature is most conveniently indicated by introducing an exponent n to V in equation 3, obtaining

$$P_1V_1^n = P_2V_2^n = P_3V_3^n = K \quad \text{(a constant)} \qquad (7)$$

in which the subscripts refer to different conditions of the same weight of gas.

It has been found by experiment that the majority of compression and expansion curves will follow the relationships expressed by equation 7. The value of the exponent n usually lies between 1.0 and 1.5, depending upon a number of factors, such as the peculiarities of the gas compressed, the specific heats of the gas, the degree of cooling, the operating characteristics of the compressor cylinder, or the amount of ring leakage.

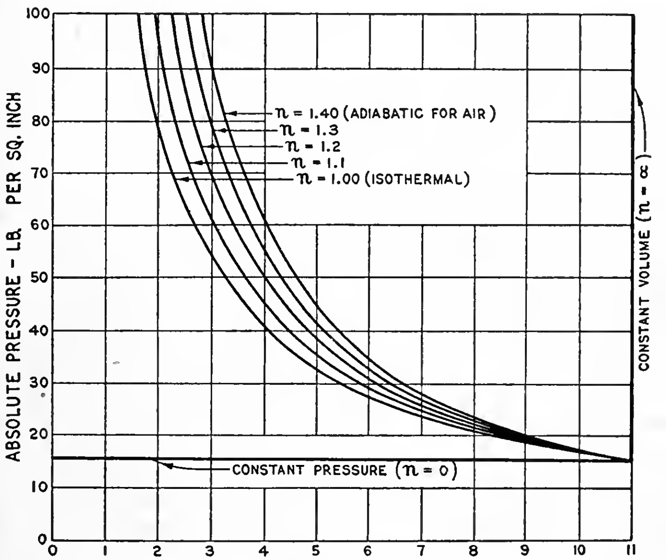


FIG. 5.

Figure 5 shows various compression curves for a compressor with approximately atmospheric intake, which discharges at 100 lb. per sq. in. abs. The effect of changes of the value of n is clearly indicated. If these curves are plotted on logarithmic paper, as in Fig. 6, they then become straight lines, with slopes equal to the tangents of the angles A, B, C, etc. The isothermal has a slope of 1.0 and is readily recognized as an equilateral hyperbola. The slopes of the other curves are usually greater than 1.0.

Equation 7 may be taken as the type formula for the expansion and compression of any gas. When the exponent $n = 1.00$, the change is said to be isothermal, or the pressure-volume change has taken place without any change in temperature. If n is greater than 1.00, the expansion is said to be polytropic. There is a special value of n, known as k or γ, which is the adiabatic for the particular gas concerned. In adiabatic expansions, no heat is taken up or given off, and all changes occur at the expense of the intrinsic energy of the gas itself. No change of entropy takes place during an adiabatic compression or expansion.

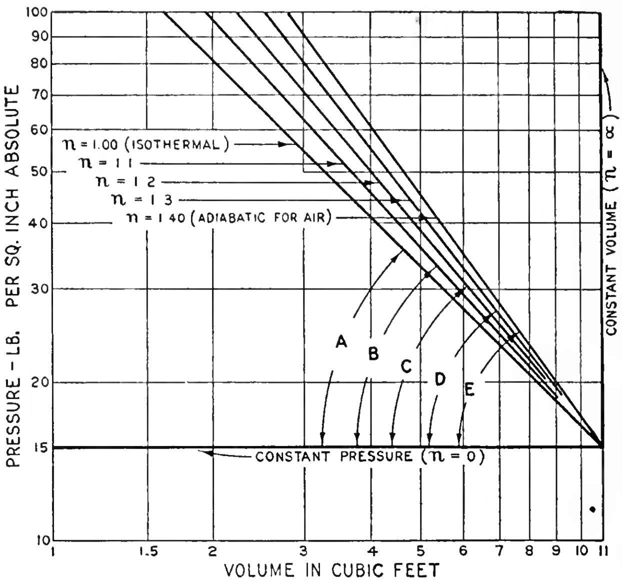


Fig. 6.

The value of n for adiabatic compression is the ratio of the specific heats of the gas compressed (C_p/C_v), where C_p is the specific heat at constant pressure and C_v is the specific heat at constant volume. The value of n for adiabatic compression of air is usually taken as 1.40 at 1 standard atmosphere and 15°C. For other gases, the adiabatic value of n is usually somewhat less than for air.

PROBLEMS

1. Reduce 187 cu. ft. of methane, at a pressure of 255 lb. per sq. in. abs., to standard cubic feet at 14.73 lb. per sq. in. abs. *Ans.:* 3237.27 standard cubic feet. Table III, being based on a local atmospheric pressure of 14.6 instead of

14.73 lb. per sq. in., may be used with a slight inaccuracy: 255 lb. abs. − 14.6 = 240.4 lb. per sq. in. gauge. By interpolation between 240 and 241, get a factor of 17.308. Multiply 187 × 17.308 = 3236.6 standard cubic feet, which is sufficiently accurate in the majority of cases. See Chapter XIII.

2. A 3-cu. ft. cylinder of oxygen contains 15 lb. of the gas at a temperature of 86°F. The specific gravity of the oxygen is 1.105. Atmospheric pressure is assumed to be 14.6 lb. per sq. in. abs. What is the pressure in the cylinder, as shown on a gauge? How many standard cubic feet at 14.73 lb. per sq. in. abs. and 60°F. does it hold? *Ans.:* 900 lb. per sq. in. gauge; 186.3 standard cubic feet. For deviation correction, see Chapter XIII.

3. An air-tank of 50-cu. ft. capacity will rupture when the pressure reaches 600 lb. per sq. in. gauge. The barometer reads 29.72 in. of mercury. How many standard cubic feet of air must be pumped into the tank before it will burst, if we disregard the effect of temperature? *Ans.:* 2086 standard cubic feet.

4. The gas in a 10-cu. ft. tank weighs 9.6 lb. and is under a pressure of 240 lb. per sq. in. gauge. The temperature is 79°F., and the barometer is 28.6 in. of mercury. What is the specific gravity of the gas? *Ans.:* 0.754.

5. How many cubic feet will 5 standard cubic feet of natural gas occupy at 8 in. of mercury vacuum, if the atmospheric pressure is 14.6 lb. per sq. in. abs.? *Ans.:* 6.9 cu. ft. Using Table III, the factor for 8-in. vacuum is 0.724. Divide 5 by 0.724 = 6.90 cu. ft.

CHAPTER IV

ENERGY RELATIONS — SINGLE-STAGE COMPRESSION

A pressure-volume diagram of a typical compression cycle is shown in Fig. 7. Gas is taken in at A at pressure P_1 and volume V_1. The gas is then compressed along the line AB, the discharge valve opening at B when the pressure P_2 is reached. At point C, the discharge valve closes and the piston begins its return stroke, allowing the pressure to

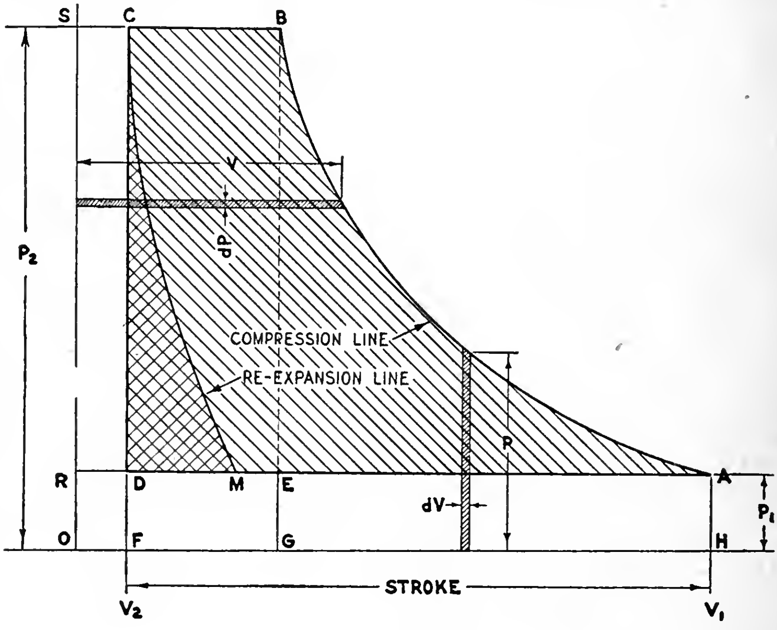


FIG. 7.

fall again to P_1 along the line CM. When the intake pressure P_1 is again reached, the intake valve opens. Intake occurs along the line MA.

Work may be defined as the product of force by distance, or of pressure by volume. In Fig. 7, the work of compression is represented by the shaded area $ABCM$.

To evaluate the work done along the compression line AB, let $dW = V\,dP$.

From equation 7, $P = K/V^n$. By raising both sides of this equation to the $1/n$ power, we get

$$P_1^{1/n}\,V_1 = K^{1/n}$$

or

$$V = \left(\frac{K}{P}\right)^{1/n}$$

Substituting

$$W = K^{1/n}\int_1^2 \frac{dP}{P^{1/n}} \tag{8}$$

The integration of equation 8 gives two different results, according to whether or not the value of n is equal to unity. For isothermal compression, when $n = 1.00$, equation 8 becomes

$$W = K\int_1^2 \frac{dP}{P}$$

which, after integration, reduces to

$$W = -K\,\log_\epsilon\left(\frac{P_2}{P_1}\right) + C \tag{9}$$

The constant of integration C is found to be zero when $P_1 = P_2$ and when no work is performed. The negative sign before K may be disregarded, as it merely shows that external work must be added to the compression cycle.

For P_2/P_1 we may substitute the term R, better known as the ratio of compression. Transferring from natural to common logarithms, $\log_\epsilon R = 2.302 \log_{10} R$. When $n = 1.00$, $K = P_1V_1$. In ordinary compressor practice, V_1 is usually taken in standard cubic feet, measured at 14.73 lb. per sq. in. abs. (30 in. of mercury) and 60°F. The term P_1V_1 then becomes 14.73 V_1, which must be multiplied by 144 to reduce to pounds per square foot. Equation 9 then becomes

$$W = 144 \times 14.73\,V_1 \times 2.302 \log R$$

$$W = 4883\,V_1 \log R \tag{10}$$

Equation 10 gives the work from zero volume to the line AB, and includes the area $ABSR$. In the meantime, re-expansion is taking place on the other side of the cylinder along the line CM, returning to the cycle the work area $CMRS$, which is of opposite sign from the total area $ABSR$. The net work is therefore the difference in areas, or $ABCM$. In arriving at this conclusion, we must assume that the

exponent of compression n is the same along AB as it is along CM, which is not always true. See equation 59.

Equation 10 gives results in foot-pounds. To reduce to horsepower, it is necessary to introduce a rate of doing work into the formula. This may be done by taking V_1 in standard cubic feet per minute, and dividing by 33,000 ft-lb. per min., the equivalent of 1 hp. Equation 10 then becomes

$$HP = \frac{4883\ V_1 \log R}{33{,}000}$$

$$HP = 0.1479\ V_1 \log R \tag{11}$$

in which V_1 is expressed in standard cubic feet per minute.[1] To obtain units of 1000 cu. ft. per 24 hours (abbreviated M.c.f.) which are used in the natural-gas industry, equation 11 becomes

$$HP = \frac{0.1479\ V_1 \log R \times 1000}{24 \times 60}$$

$$HP = 0.10275\ V_1 \log R \tag{12}$$

in which V_1 is expressed in thousands of cubic feet per 24 hours.[2]

Isothermal compression is seldom attained in practice, but the concept is useful as the ideal to which all compressor performance may be compared.

To compute the work done in polytropic performance, it is necessary to integrate equation 8 for values of $n \neq 1$, obtaining

$$W = -\frac{n}{n-1} K^{1/n} [P_2{}^{(n-1)/n} - P_1{}^{(n-1)/n}]$$

$$= -\frac{n}{n-1} P_1{}^{1/n} V_1 [P_2{}^{(n-1)/n} - P_1{}^{(n-1)/n}] \tag{13}$$

As above, the minus sign before $n/(n-1)$ may be disregarded, as it merely shows that external energy must be supplied to the cycle.

Multiplying both numerator and denominator of the right-hand side of the above equation by $P_1{}^{(n-1)/n}$ we obtain

$$W = \frac{n}{n-1} V_1 P_1{}^{1/n} P_1{}^{(n-1)/n} \left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right]$$

The ratio of compression $R = P_2/P_1$. Substituting, we obtain

$$\boxed{W = \frac{n}{n-1} P_1 V_1 [R^{(n-1)/n} - 1] \text{ ft-lb.}} \tag{14}$$

[1] See equation 79.

[2] See equation 80.

Taking the intake gas in standard cubic feet at 14.73 lb. per sq. in. abs., so that $P_1 = 14.73$, then

$$W = \frac{n}{n-1} \quad 144 \times 14.73\, V_1\, [R^{(n-1)/n} - 1]$$

$$W = \frac{n}{n-1}\, 2121\, V_1\, [R^{(n-1)/n} - 1] \text{ ft-lb.} \tag{15}$$

To get this equation in terms of horsepower, take V_1 in standard cubic feet of gas handled per minute, and divide by 33,000 ft-lb. per min., obtaining

$$HP = \frac{n}{n-1}\,\frac{2121}{33{,}000}\, V_1\, [R^{(n-1)/n} - 1]$$

$$= \frac{n}{n-1}\, 0.0643\, V_1\, [R^{(n-1)/n} - 1] \tag{16}$$

in which V_1 is taken in standard cubic feet per minute.[1]

To get horsepower in terms of thousands of standard cubic feet per 24 hours, multiply equation 16 by $1000/(24 \times 60)$ and obtain

$$HP = 0.0446\, V_1\, \frac{n}{n-1}\, [R^{(n-1)/n} - 1] \tag{17}$$

in which V_1 is taken in thousands of standard cubic feet per 24 hours (M.c.f.).[2]

Equations 16 and 17 give the theoretical horsepower required for polytropic compression, making no allowance for compressor efficiency, valve losses, or deviation from the gas laws. When $n = C_p/C_v$, the specific-heat ratio of the gas being compressed, these formulas give the theoretical adiabatic horsepower.

Figure 8 illustrates the comparative power required to compress 10 cu. ft. of air from 15 lb. to 100 lb. per sq. in. abs. at different values of n. A clearance volume of 1 cu. ft. is assumed. The work of isothermal compression is shown on the diagram as the area $ABGHA$. For $n = 1.1$, the work is represented by $ACGHA$. When $n = 1.4$, the adiabatic value for air, the work is $AFGHA$. Isothermal air compression, therefore, shows a saving in work of the area $AFBA$ when compared with adiabatic compression. Owing to radiation, jacket cooling, condensation, and other causes, few compressors ever operate adiabatically. As it is practically impossible to keep the gases at the same temperature during compression, isothermal compression is also rare.

[1] See equation 81.

[2] See equation 82.

Figure 8 shows that the area of the indicator card, and therefore the power required, becomes smaller as the value of n is decreased. Thus it is evident that, the more heat that is removed from the compressor during the cycle, the more efficient the compression becomes and the more closely it approaches the isothermal. For this reason, compressor

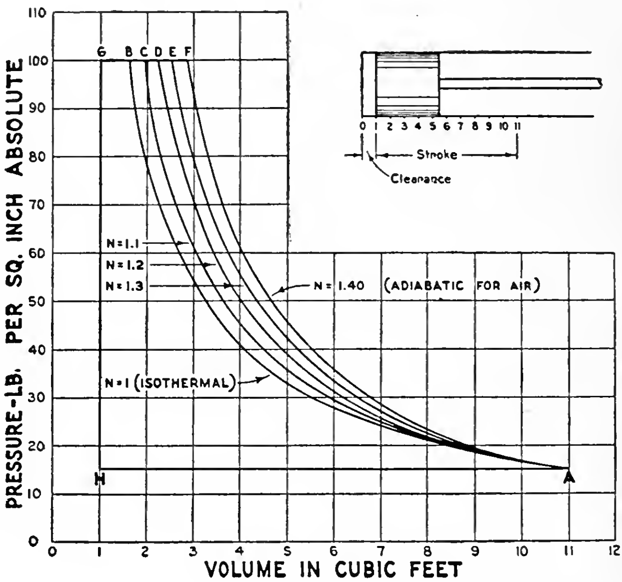


FIG. 8.

cylinders are water-jacketed, and intercoolers are provided between stages in multistage compression, in order to bring the operation as near the isothermal as possible. Theoretically, actual compression curves should lie somewhere between the isothermal and the adiabatic, usually closer to the latter.

Table IV gives factors for the solution of equations 11 and 16. In the column marked "isothermal," values of 0.1479 log R are given. In the other columns are values from equation 16, for a volume of 1 cu. ft. per min.

To obtain the theoretical horsepower of compression, multiply the proper factor in Table IV by the initial volume of gas to be compressed, taken in standard cubic feet per minute.

Table V gives factors for the solution of equations 12 and 17. These factors are similar to those in Table IV, except that the initial volumes are to be taken in thousands of standard cubic feet per 24 hours.

PROBLEMS

1. An air compressor is required to supply 9000 standard cubic feet per hour to an air lift at 73 lb. per sq. in. gauge. Atmospheric pressure 14.6 lb. per sq. in. abs. What horsepower is required for isothermal conditions, assuming 80 per cent overall efficiency? What horsepower is required for adiabatic compression, taking $n = 1.4$? *Solution:* 9000 cu. ft. per hr. = 150 cu. ft. per min. Ratio of compression = $(73 + 14.6)/14.6 = 6.0$. In Table IV, find a factor of 0.1152 for isothermal compression and a ratio of compression of 6.0. $HP = (150 \times 0.1152)/0.80 = 21.6$ hp. For adiabatic compression, $HP = 28.25$.

2. What is the theoretical isothermal horsepower required to compress 500 M.c.f. of natural gas from 14.7 to 73.5 lb. per sq. in. abs.? *Solution:* Ratio of compression is $73.5/14.7 = 5.0$. Factor from Table V is 0.0718. $HP = 500 \times 0.0718 = 35.9$.

CHAPTER V

COMPRESSOR CAPACITY

In computing the quantity of gas handled by a compressor, we must distinguish carefully between the terms: *cubic feet, cubic feet at a given pressure, standard cubic feet, nominal compressor displacement, volumetric displacement,* and *compressor capacity.* These terms, though similar, are by no means synonymous, and must be correctly used to avoid confusion.

It is evident that the term, " 10 cu. ft. of gas " is meaningless unless the pressure of the gas is also stated. We may speak of 10 cu. ft. at 50 lb. per sq. in. gauge, however, and be much more definite. But to give a complete picture of the quantity of gas referred to it is necessary to state the volume, temperature, and pressure, all of which are to be measured at the same point and at the same time. In order to avoid confusion and to conform to industrial usage, all volumes in this book will be based on the standard cubic foot, measured at 30 in. of mercury (14.73 lb. per sq. in. abs.) and 60°F., unless otherwise stated.

The nominal displacement of a compressor is a function of the bore and stroke, and of the speed in revolutions per minute. It is simply the space displaced by the piston, making no allowance for re-expansion or clearance. A double-acting compressor of 12-in. stroke having a piston diameter of 9.6 in. will have a nominal displacement of approximately 1 cu. ft. per revolution. It is *not* correct to speak of displacements in *standard* cubic feet, as we are not concerned with temperatures and pressures, but with volumes only.

The volumetric displacement of a compressor is obtained by multiplying the nominal displacement by the volumetric efficiency.[1] If the compressor mentioned in the last paragraph operates at 81 per cent volumetric efficiency, the volumetric displacement will be $1.0 \times 0.81 = 0.81$ cu. ft. per revolution.

Capacity, when referred to a compressor, is taken to mean the actual amount of gas handled by the machine under standard conditions. If the intake pressure is 14.73 lb. per sq. in., and the intake temperature is 60°F., the capacity and the volumetric displacement will be practically the same. This condition would be nearly true for air compressors operating near sea level and having an intake temperature of 60°F.

[1] For explanation of volumetric efficiency, see Chapter VI.

As the intakes of many compressors are not at atmosphere, it is obvious that the term capacity expresses merely the amount of gas compressed, reduced to standard cubic feet.

If the intake on the 9.6-in. cylinder mentioned above were at 60°F. and 59 lb. per sq. in. abs., the capacity would be $(59/14.73) \times 0.81 = 3.24$ standard cubic feet per revolution for 81 per cent volumetric efficiency.

As air compressors must take air from the atmosphere, it is evident that their capacities will be affected by the temperature of the air and also by its pressure, which is a function of the altitude of the compressor above sea level. From equation 3, PV/T at any point $= P_sV_s/T_s$, where the subscript s refers to standard conditions of 14.73 lb. per sq. in. and 60°F.

The capacity factor for air compressors may be expressed as

$$F = \frac{V_s}{V} = \frac{T_sP}{P_sT} = \frac{520\,P}{14.73\,T}$$

$$= \frac{35.3\,P}{T} \tag{18}$$

For accuracy, the temperatures and pressures should be taken as close to the intake valves as possible.

Table II gives a few values of the air intake capacity factor for various altitudes and temperatures. This table shows that the capacity of an air compressor decreases with increase of altitude and also with increase of temperature.

The volumetric displacement of any compressor may be taken as

$$D_v = \frac{2\,\pi}{4 \times 1728}\,d^2\,L\,N\,E$$

$$= 0.000909\,d^2\,L\,N\,E \tag{19}$$

where D_v is the volumetric displacement of the compressor in cubic feet per minute; d is the diameter of the low-pressure cylinder in inches; L is the stroke of the low-pressure cylinder in inches; N is revolutions per minute of the compressor; and E is the volumetric efficiency of the low-pressure cylinder as a decimal fraction. The compressor is assumed to be double acting.

For displacements in thousands of cubic feet per 24 hours:

$$D_v = \frac{2\,\pi \times 24 \times 60}{4 \times 1000 \times 1728}\,d^2\,L\,N\,E$$

$$= 0.001309\,d^2\,L\,N\,E \tag{20}$$

where D_v is the volumetric displacement of the compressor in thousands of cubic feet per 24 hours.

Nominal displacements may be computed by omitting E from equations 19 and 20.

Table VI gives nominal compressor displacement factors based on compressors of 20-in. stroke running at 200 r.p.m. Values for several different sizes of piston rods are given at the bottom of the table. To secure the desired nominal displacement, subtract the figure given under the rod size from the value under the cylinder size. All compressors are considered to be double acting.

Table VII gives revolutions per minute and stroke correction factors, to be used with Table VI.

To secure the capacity of a compressor in standard cubic feet, first compute the volumetric displacement from equation 19 or 20. The capacity then is

$$C = \frac{D_v(P_g + P_a)}{14.73} \tag{21}$$

where C is the capacity[1] in standard cubic feet; D_v is the volumetric displacement; P_g is the intake pressure in pounds per square inch gauge; and P_a is the local barometric pressure in pounds per square inch. Table III contains the factor $(14.6 + P_a)/14.73$. Multiply the factors in this table by volumetric displacements to secure capacities. The atmospheric pressure of 14.6 lb. per sq. in. contained in these factors will probably be accurate enough for altitudes up to 350 ft. above sea level.

For air compressors, where the intake is atmospheric and a temperature factor is required,

$$\begin{aligned} C &= \frac{D_v \times P_a \times T_s}{14.73\,T} \\ &= \frac{D_v \times 520\,P_a}{14.73\,T} \\ &= \frac{D_v \times 35.3\,P_a}{T} \end{aligned} \tag{22}$$

Equation 22 may also be solved by multiplying volumetric displacements by the altitude-temperature factors in Table II.

PROBLEMS

1. A compressor has a $12\frac{1}{2}$-in. cylinder and an 18-in. stroke, and runs at 250 r.p.m. at an elevation of 295 ft. above sea level. The volumetric efficiency is 85 per cent, and the intake pressure is 47 lb. per sq. in. gauge. Piston-rod

[1] See equations 76, 77, and 78.

diameter is $1\frac{3}{4}$ in. Required: the nominal displacement, the volumetric displacement, and the capacity in thousands of cubic feet per 24 hours. *Solution:* Using equation 20, nominal displacement is 919 M.c.f., less 9 M.c.f. for rod, making 910 M.c.f. net. Volumetric displacement is $0.85 \times 910 = 774$ M.c.f. From Table III get a pressure factor of 4.181 for 47 lb. gauge. Capacity is $4.181 \times 774 = 3238$ M.c.f. Using Table VI: factor for $12\frac{1}{2}$-in. cylinder and $1\frac{3}{4}$-in. rod is $818.2 - 8.0 = 810.2$. From Table VII get a factor of 1.125 for 250 r.p.m. and 18-in. stroke. Nominal displacement is $810.2 \times 1.125 = 910$ M.c.f. Volumetric displacement is $910 \times 0.85 = 774$ M.c.f. Capacity is $4.181 \times 774 = 3238$ M.c.f.

2. What size cylinder should be selected to compress 1,000,000 cu. ft. of gas (1000 M.c.f.), assuming a 2-in. rod, 190 r.p.m., 88 per cent volumetric efficiency, and 16-in. stroke? Intake to be at 20 lb. per sq. in. abs. *Solution:* Divide capacity of 1000 M.c.f. by pressure ratio (20/14.73), and volumetric efficiency. Nominal displacement is 837.5 M.c.f. Use equation 20 with E omitted. Disregarding rod, $d = 14.5$ in. Check by using Table VI: nominal displacement for $14\frac{1}{2}$-in. cylinder is 1100.9; for 2-in. rod is 10.5; net value is 1090.4 M.c.f. Multiply by stroke and r.p.m. factor from Table VII: $1090.4 \times 0.760 = 828$ M.c.f. Capacity is $828 \times (20/14.73) \times 0.88 = 978$ standard M.c.f. Values for 15-in. cylinder: net nominal displacement, 1167.6 M.c.f. $\times\ 0.760 = 887$ M.c.f. Capacity is $887 \times 20/14.73 \times 0.88 = 1060$ M.c.f.

CHAPTER VI

VOLUMETRIC EFFICIENCY AND CLEARANCE

The volumetric efficiency E which occurs in equations 19 and 20 is the ratio of the volume of gas actually compressed to the theoretical compressor capacity.[1] Thus, if a compressor running at a given speed has a nominal displacement of 100 cu. ft. per min., with intake at 24 lb. gauge, the theoretical capacity would be 2.62 (see Table III) $\times$ 100 = 262 standard cubic feet per minute. If the machine actually handles only 212 standard cubic feet per minute, the volumetric efficiency would be 212/262 = 81 per cent, provided that there is no deviation[1] from the ideal gas laws.

Volumetric efficiency is a function of the clearance of the low-pressure cylinder and the ratio of compression. Clearance may be defined as the volume remaining in the cylinder at the extreme position of the piston (dead center), divided by the displacement of the cylinder. As the clearance volume is filled with gas that has been compressed, this gas expands again as the piston moves outward, thus delaying the opening of the intake valves and consequently the amount of gas that is taken into the cylinder on the next stroke. The intake valves cannot open until the gas in the cylinder has expanded to the intake pressure; consequently the effect of clearance is to reduce the capacity of the compressor.

Modern types of valves make it impossible to build a compressor without a fair amount of clearance. The principal effect of clearance, however, is simply to increase the size of the cylinder required, without noticeably increasing the horsepower. Equations 11, 12, 16, and 17, show that the theoretical horsepower needed to compress a given amount of gas depends solely upon the volume of gas handled, together with the ratio of compression, and not on the clearance.

Figure 9 is an indicator card, or pressure-volume record, of an air compressor taking in air along the line EE. Compression takes place from A to B. The discharge valve opens at B, and air is discharged from B to C. At the end of the stroke, at the point C, the clearance volume of 1 cu. ft. of air at 100 lb. per sq. in. abs. still remains in the cylinder.

[1] See Chapter XIII.

As the piston moves out, this air expands along the line CD until it reaches the intake pressure line EE at D, at which point the intake valve opens and the intake begins. Air is taken in along the line DA. Graphically, the volumetric efficiency is the ratio of $\overline{DA}$ to $\overline{HA}$.

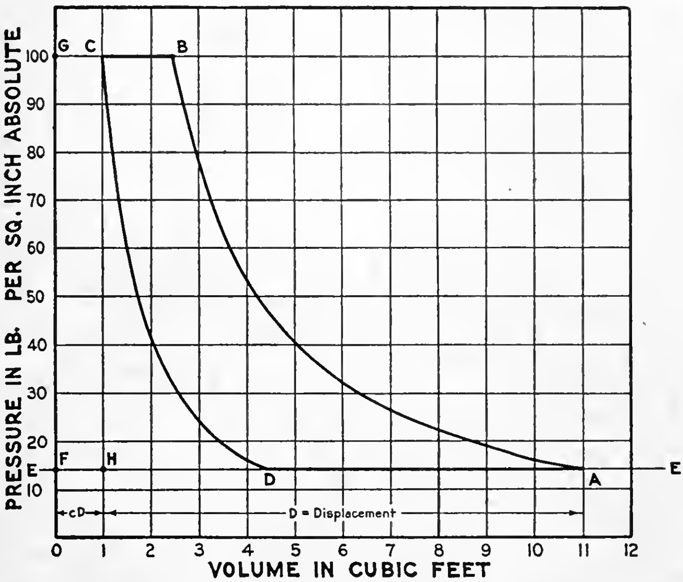


FIG. 9.

In order to get a relation between volumetric efficiency and clearance, let D represent the displacement per stroke. The clearance volume is then cD, when c is taken as a decimal. The volume of gas compressed at each stroke is therefore $D + cD = (1 + c)\,D$. From this must be subtracted the volume occupied by cD after expansion. Using equation 7 and assuming polytropic expansion,

$$PV^n = P_C\,V_C{}^n = P_D\,V_D{}^n$$

where the subscripts C and D refer to conditions at these points on Fig. 9. As assumed, $V_C = cD$. The final volume of cD after expansion,

$$V_D = \left(\frac{P_C}{P_D}\right)^{1/n} cD = R^{1/n}cD$$

Thus the gas which occupied the volume GC at 100 lb. will have a volume FD at pressure EE. As volumetric efficiency is the ratio of the

actual intake volume to the displacement,

$$E_v = \frac{D + cD - R^{1/n}cD}{D}$$

$$= 1 + c\,(1 - R^{1/n})$$

or, as it is usually given,[1]

$$E_v = 1 - c\,(R^{1/n} - 1) \tag{23}$$

Table VIII gives values of $[R^{1/n} - 1]$ for the solution of equations 23 and 60.

Maximum Ratio of Compression. A special case of volumetric efficiency occurs when the clearance is so large and the ratio of com-

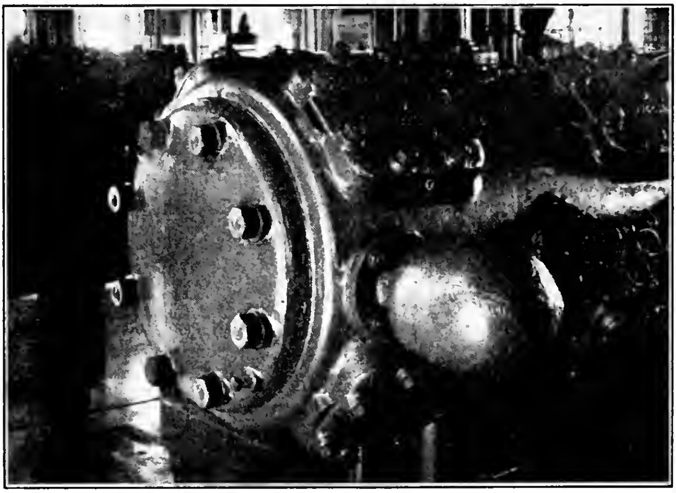


FIG. 10. Simple form of clearance pocket on Clark compressor cylinder.

pression is so high that re-expansion takes place during the entire intake stroke. The result is that the intake valve does not open, and consequently no gas is taken in or discharged from the compressor. The volumetric efficiency under these conditions is therefore zero.

By placing $E_v = 0$ in equation 23, we obtain the maximum ratio of compression

$$R_m{}^{1/n} = \left(\frac{1 + c}{c}\right)$$

$$R_m = \left(\frac{1 + c}{c}\right)^n \tag{24}$$

[1] See equations 60 and 61.

Table IX gives data for computing the maximum ratio of compression when n and c are known.

For further discussion of volumetric efficiency see equation 61, Chapter XI.

PROBLEMS

1. What is the volumetric efficiency of an air compressor with intake at 26.5 in. of mercury and discharge at 65 lb. per sq. in. gauge, if the clearance is 10 per cent? The exponent of compression n may be taken as 1.30. How many clearance pockets (see Fig. 10) which increase the cylinder clearance by 5 per cent each may be opened into the cylinder before the compressor ceases to discharge air? *Solution:*. Atmospheric pressure is $26.5 \times 0.4911 = 13$ lb. per sq. in. abs. Ratio of compression is $(65 + 13)/13 = 6$. Volumetric efficiency is 70.3 per cent. Substituting in equation 24, maximum clearance for ratio of compression of 6 is 33.7 per cent. Therefore four clearance pockets may be opened into the cylinder before discharge ceases.

2. What is the maximum percentage of clearance for a ratio of compression of 16, assuming an air compressor operating at $n = 1.40$. *Ans.:* See Table IX. Maximum clearance is 16 per cent.

3. What is the volumetric efficiency of a compressor operating on wet gas with a compression exponent of $n = 1.1$, having 8 per cent clearance and a ratio of compression of 2.8? *Ans.:* 87.6 per cent.

CHAPTER VII

INDICATED HORSEPOWER

The engine indicator mechanism (see Fig. 11) produces a card which shows the pressure in the compressor cylinder throughout the stroke. When the cylinder clearance is known, such a card can be used to draw a complete pressure-volume diagram similar to Fig. 9.

The fundamental relation of the indicator card may be expressed:

$$IHP = \frac{P\,L\,A\,N}{33{,}000} \tag{25}$$

where IHP = indicated horsepower actually developed in the compressor cylinder.

P = mean effective pressure throughout the stroke (MEP) in pounds per square inch.

L = stroke in feet.

A = cross-sectional area of cylinder in square inches. If the compressor is double-acting, multiply the area of the bore by 2 and subtract the area of the piston rod.

N = revolutions per minute of the machine (r.p.m.).

Values of $LAN/33{,}000$ are given in Table VI for various sizes of cylinders and rods for double-acting compressors. Table VII gives stroke and r.p.m. factors. For numerical results, it only remains to evaluate P, the mean effective pressure, from the indicator card.

The area of the card in square inches should first be obtained by means of a planimeter. If no planimeter is available, a fairly reliable result may be obtained by superimposing a sheet of transparent squared paper over the indicator card, and counting the squares as carefully as possible. Another method is to erect equally spaced ordinates normal to the atmospheric line, the area being found by the trapezoidal or Simpson's rule.

While the indicator card is being taken, care should be exercised to include the "atmospheric line," which will be drawn by the indicator mechanism while the compressor is in motion, with the pressure element disconnected from the cylinder. The atmospheric line will serve to orient the card, as the line of zero pressure must be parallel to it.

After the card is taken from the indicator drum, perpendiculars should be erected to the atmospheric line in such a manner that they touch the

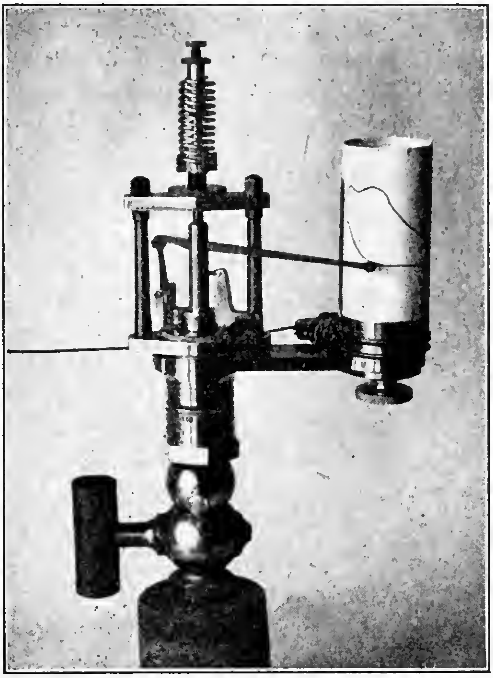


FIG. 11. Crosby engine indicator.

extreme right and left edges of the closed figure traced by the stylus. The distance along the atmospheric line between perpendiculars will then be the length of the card. See Fig. 12.

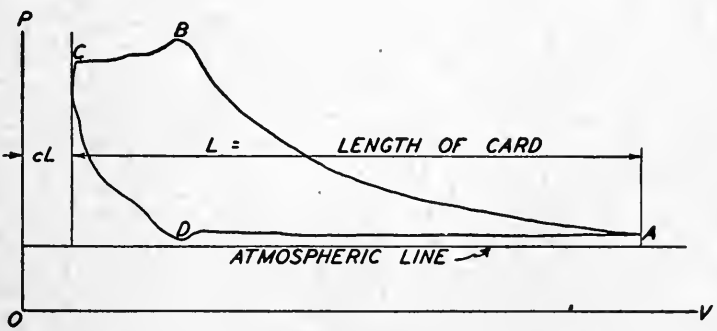


FIG. 12. Typical compressor indicator card.

The average or mean effective pressure will then be

$$MEP = \frac{a\,S}{l} \tag{26}$$

where a = area of the indicator card in square inches.
S = scale of indicator mechanism pressure spring in pounds per square inch per inch.
l = length of card in inches.

If the compressor is double-acting, indicator cards should be taken from both the head and crank ends of the cylinder. The power of each may then be computed separately. If the values of $LAN/33{,}000$ from Table VI are used, the average mean effective pressure of both ends should be taken.

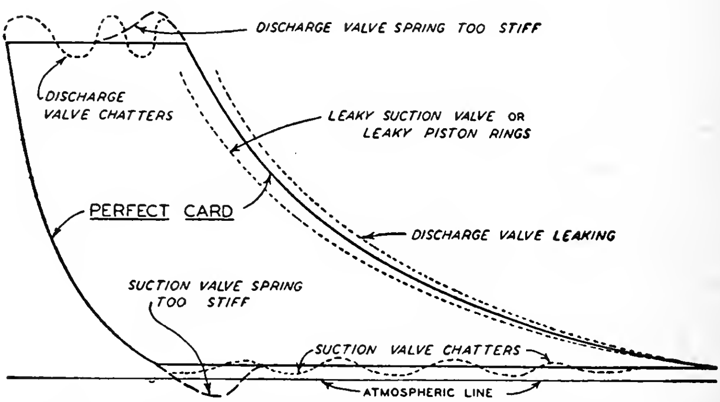


FIG. 13. Indicator card showing irregularities of compressor operation.

Figure 13 is a standard indicator card, showing some of the irregularities which may be encountered.

For computing the mean effective pressure numerically, we may define it as that pressure which, when multiplied by the piston displacement, will give the work inside the cylinder. The work, in foot-pounds, when divided by 33,000, will give the indicated horsepower.

Equations 10 and 15 give theoretical compressor horsepower. The term V_1 is in standard cubic feet and has already been corrected for volumetric efficiency. From the definition in the above paragraph, the mean effective pressure would be composed of all the remainder of equations 10 and 15, after taking out the term for volume.

For isothermal compression,[1] from equation 10,

$$\begin{aligned} MEP &= 14.73\, E_v \log_\epsilon R \\ &= 14.73\, E_v\, 2.302 \log_{10} R \\ &= 33.91\, E_v \log R \end{aligned} \tag{27}$$

For polytropic compression,[1] from equation 15,

$$MEP = 14.73 \frac{E_v\, n}{n-1} [R^{(n-1)/n} - 1] \tag{28}$$

Table X gives data for mean effective pressure as a function of the ratio of compression, for isothermal and polytropic conditions. Tabular values must be multiplied by the volumetric efficiency (E_v). All the calculations for theoretical horsepower of various-sized cylinders may be performed with data from Tables VI, VII, and X.

PROBLEMS

1. It is required to compute the indicated compressor horsepower of a compressor with a 24-in. cylinder and a 36-in. stroke, which runs at 85 r.p.m. The diameter of the piston rod is 3 in. An indicator card from the head end has an area of 4.35 sq. in., and from the crank end 4.05 sq. in. Length of the card is 3.02 in. The card was taken with a 20-lb. spring. *Solution:* From Tables VI and VII, factors for $LAN/33{,}000$ are $(9.140 - 0.071) \times 0.765 = 6.93$. Average card area is $(4.35 + 4.05)/2 = 4.20$ sq. in. $MEP = (4.2 \times 20)/3.02 = 27.8$ lb. per sq. in. $IHP = 6.93 \times 27.8 = 192.5$ hp.

2. Required, the theoretical horsepower of an air compressor running at 275 r.p.m., with 10-in. stroke and 10-in. compressor cylinder, if the barometer is 28 in. of mercury, the discharge pressure is 55 lb. per sq. in. gauge, and the volumetric efficiency is 90 per cent. *Solution:* Ratio of compression is $[55 + (0.491 \times 28)]/(0.491 \times 28) = 5.0$. $LAN/33{,}000$ from Tables VI and VII is $1.587 \times 0.6875 = 1.092$. MEP from Table X, assuming $n = 1.30$, is $28.72 \times 0.90 = 25.85$. $IHP = 1.092 \times 25.85 = 28.2$ hp. As a check, compute the problem from equation 17. From Tables VI and VII, the displacement is $523.6 \times 0.6875 = 360$ M.c.f. The capacity is $360 \times 0.90 = 324$ M.c.f. From Table V, get a horsepower factor of 0.087. Theoretical horsepower is $324 \times 0.087 = 28.2$.

3. A compressor has its intake at 15 lb. per sq. in. abs. and discharges at 60 lb. abs. The clearance is 10 per cent. Determine the mean effective pressure by computation and also by graphical means. Assume $n = 1.2$. *Solution:* On a 2-cycle × 2-cycle logarithmic cross-section sheet, plot the indicator card by assuming a displacement of 10 cu. ft. and a clearance volume of 1 ft. The beginning of the compression stroke will be at 15 lb. abs. and a volume of $10 + 1 = 11$ cu. ft. It will extend upward towards the left, with a slope of 1.2, meet-

[1] See Chapter XIII.

ing the 60-lb. pressure line at approximately 3.5 cu. ft. The re-expansion line begins at a volume of 1 and a pressure of 60 lb., and extends downward to the right, parallel to the compression line, and intersecting the suction pressure line of 15 lb. abs. at approximately 3.15 cu. ft. By analogy with Fig. 9, $\overline{DA}/\overline{HA}$ = 78.5 per cent, which is the volumetric efficiency. Now plot the card on uniform cross-section paper, with abscissas of 1 in. = 1 cu. ft. and ordinates of 1 in. = 10 lb. pressure. The area of this figure should be approximately 18 sq. in. The mean effective pressure is $(18 \times 10)/10 = 18$ lb. per sq. in., which is seen to be the average height of the card in pounds per square inch per inch. Check the volumetric efficiency from Table VIII, and get a value of 78.25 per cent. Now consult Table X, and get the factor for a ratio of compression of 4 and $n = 1.2$. The mean effective pressure is $22.98 \times 0.7825 = 18$ lb. per sq. in.

CHAPTER VIII

TWO-STAGE COMPRESSION

When the ratio of compression exceeds 4 or 5, the temperature rise of the gas being compressed becomes excessive, and much heat, and consequently power, is lost which otherwise might be saved. To avoid such losses, it is common practice to divide the compression into several stages, with intercooling between stages. In this manner the overall compression curve will more nearly approach the ideal, or isothermal, and there will at the same time be a considerable saving of power, to say nothing of decreased strain and wear on engine and compressor parts and bearing surfaces.

Temperature Rise. An expression of the temperature rise of gas during compression may be obtained by eliminating V between equations 4 and 7.

From equation 4,

$$\frac{V_1}{V_2} = \frac{P_2 T_1}{P_1 T_2}$$

From equation 7,

$$\frac{V_1^n}{V_2^n} = \frac{P_2}{P_1}$$

$$\boxed{\frac{V_1}{V_2} = \left(\frac{P_2}{P_1}\right)^{1/n}}$$

Equating,

$$\left(\frac{P_2}{P_1}\right)^{1/n} = \left(\frac{P_2}{P_1}\right)\left(\frac{T_1}{T_2}\right)$$

$$\boxed{\left(\frac{P_2}{P_1}\right)^{(1-n)/n} = \frac{T_1}{T_2}}$$

For P_2/P_1 we may substitute R, the ratio of compression, obtaining

$$T_2 = T_1 R^{(n-1)/n} \tag{29}$$

Values of $R^{(n-1)/n}$ are given in Table XI.

As an example of the effect of intercooling between stages, refer to Fig. 14, which represents an air compressor having suction at atmosphere and discharging at 100 lb. per sq. in. abs. The compression ratio is

100/14.73 = 6.79. Using an initial temperature of 80°F. and a value of $n = 1.4$, we substitute in equation 29 and get a final temperature of 473°F.

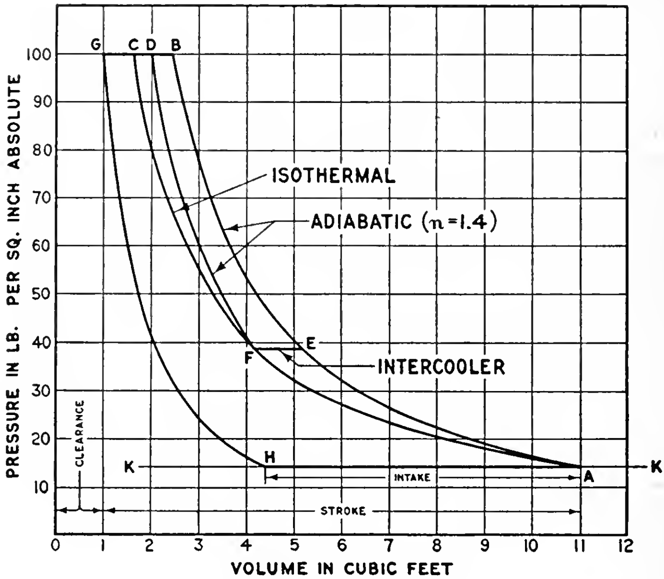


FIG. 14.

Intercooler Pressure. Let us split this compression into two stages. In order to divide the work equally between the two stages, each stage should have the same compression ratio. As the overall ratio of compression from atmosphere to 100 lb. per sq. in. abs. is 6.79, the value of R for each stage is $\sqrt{6.79} = 2.605$. The intermediate pressure would then be $2.605 \times 14.73 = 38.4$ lb. per sq. in. abs. As a check, we note that the second stage will therefore have a compression ratio of $R = 100/38.4 = 2.605$. This leads to the equation

$$P_2 = P_1\sqrt{\left(\frac{P_2}{P_1}\right)\left(\frac{P_3}{P_2}\right)} = P_1\sqrt{\frac{P_3}{P_1}} = \sqrt{P_1P_3} = P_1\sqrt{R_t} \qquad (30)$$

where P_1 = low-pressure intake in pounds per square inch absolute.
P_2 = low-pressure discharge, intercooler, and second stage intake, in pounds per square inch absolute.
P_3 = high-pressure discharge in pounds per square inch absolute.
R_t = overall ratio of compression.

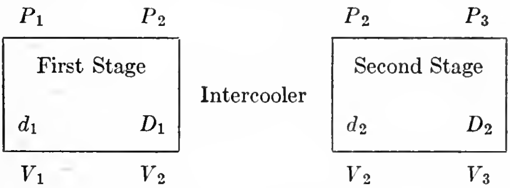
Equation 30 assumes that both high- and low-pressure cylinders have the same volumetric efficiency.

A further examination of Fig. 14 will show the advantages of two-stage compression over compression in a single step. We recall that gas compressed from 80° and atmospheric pressure at A will have a temperature of 473° when it reaches 100 lb. per sq. in. abs. at B. If the compression is limited to 38.4 lb. abs. in the first stage, the temperature will rise to only 250° at E. The gas may then be cooled to 80° at F and finally compressed to 100 lb. abs. along the line FD. The temperature at D will also be 250°, as the ratio of compression and initial temperature are both the same as before. The area $EFDB$ represents the saving in power of two-stage over single-stage compression. The line AC represents isothermal compression, beginning at the point A. It is therefore evident that, the larger the number of stages employed, the closer the actual compression line will approach the ideal or isothermal line AC.

For two-stage compression, the temperature rise may be figured for each stage separately by equation 29. But as the temperature depends solely upon the ratio of compression and the value of n, and as the ratio of each stage should be the square root of the overall ratio, then

$$\begin{aligned} T_2 &= T_1 \sqrt{R_t^{(n-1)/n}} \\ &= T_1 R_t^{(n-1)/2n} \end{aligned} \tag{31}$$

where R_t is the overall ratio of compression. This equation assumes that there is perfect intercooling, which means that the gas is to be cooled between stages so that the intake to the high-pressure cylinder is at the same temperature as that of the low-pressure intake.



Cylinder Sizes. To determine cylinder sizes, we recall from equation 3 that $P_1V_1 = P_2V_2$. Then

$$\frac{P_2}{P_1} = \frac{V_1}{V_2} = \frac{D_1 E_{v_1}}{D_2 E_{v_2}}$$

where D is the displacement of the cylinder.

E_v is the volumetric efficiency.

Displacements are directly proportional to the squares of the cylinder diameters. The term P_2/P_1 is readily recognized as the ratio of compression in the first stage of compression, and is equal numerically to

the square root of the overall ratio of compression. Assuming the same volumetric efficiency in both cylinders, then

$$R_1 = \frac{D_1}{D_2} = \frac{d_1^2}{d_2^2}$$

Using the overall ratio of compression,

$$\sqrt[4]{R_t} = \frac{d_1}{d_2} \tag{32}$$

Equation 32 assumes that there is perfect intercooling and that the stroke of both cylinders is the same.

It is sometimes desirable to figure a little closer on cylinder sizes than by equation 32, especially if there is a wide variation in the clearances and volumetric efficiencies of the two cylinders used, as often happens.

From the above equations,

$$R_1 = \left(\frac{d_1^2}{d_2^2}\right)\left(\frac{E_{v_1}}{E_{v_2}}\right)$$

where R_1 is the ratio of compression of the low-pressure cylinder.

Substituting the expression for volumetric efficiency from equation 23,

$$R_1 = \left(\frac{d_1^2}{d_2^2}\right)\frac{1 - c_1\,[R_1^{1/n} - 1]}{1 - c_2\,[R_2^{1/n} - 1]} \tag{33}$$

This equation does not admit of a direct solution but must be solved by " cut-and-try " methods.

Let us assume air compression from atmosphere to 100 lb. per sq. in. abs. with an overall ratio of compression of 100/14.73 = 6.79. It has been decided that the low-pressure cylinder will be 10 in. in diameter, with 5 per cent clearance. For equal work in each cylinder, the ratio of compression in the first stage will be $\sqrt{6.79} = 2.605 = R_1 = R_2$. Compute the volumetric efficiency from equation 23:

$$E_{v_1} = 1 - c_1\,[R_1^{1/n} - 1] = 1.00 - 0.05\ (1.98 - 1.00) = 95.1 \text{ per cent}$$

The approximate size of the high-pressure cylinder may be found from equation 32:

$$d_2^4 = \frac{d_1^4}{R_t}$$

$$d_2 = \sqrt[4]{\frac{10^4}{6.79}} = 6.2 \text{ in.,}$$

which is the approximate diameter of the high-pressure cylinder.

Suppose that we have on hand the following cylinders which may be used for the service in question:

$6\frac{1}{2}$ in. in diameter, with $12\frac{1}{4}$ per cent clearance
$6\frac{1}{4}$ in. in diameter, with 15 per cent clearance
6 in. in diameter, with 15 per cent clearance

Substitute known values in equation 33:

$$d_2{}^2 E_{v_2} = \frac{10^2 \times 0.951}{2.605} = 36.5$$

Assuming a ratio of compression in the second stage as equal to 2.605, we find values of the volumetric efficiency E_{v_2} as follows:

$6\frac{1}{2}$-in. cylinder, volumetric efficiency = 88 per cent
$6\frac{1}{4}$-in. cylinder, volumetric efficiency = 85.3 per cent
6-in. cylinder, volumetric efficiency = 85.3 per cent

Now compute values of $d^2 E_v$ for the three cylinders: $6\frac{1}{2}$-in. cylinder, 37.2; $6\frac{1}{4}$-in. cylinder, 33.3; 6-in. cylinder, 30.7. The choice would therefore be between the $6\frac{1}{2}$-in. and the $6\frac{1}{4}$-in. Let us take the $6\frac{1}{2}$-in. one, to be on the safe side.

After having decided upon the cylinder sizes, we may substitute values for d_1 and d_2 in equation 33, but since d_2 is now 6.5 in. instead of 6.2 in. as calculated, it is evident that the ratios of compression in the two cylinders will also have to change in order for the equation to hold. Since the high-pressure cylinder is a little larger than calculated, its ratio of compression will be a little less than the value of 2.605 required for equal work in both cylinders. Consequently, a little more work will have to be done by the low-pressure cylinder, and it will therefore have a little higher ratio of compression than formerly. By the somewhat laborious process of substituting values of R_1 and R_2 in equation 33, and remembering that $R_1 \times R_2 = 6.79$, it will be found that, with the 6.5-in. cylinder in the second stage of compression, $R_1 = 2.625$, and $R_2 = 2.585$, which represents a deviation of approximately $\frac{3}{4}$ of 1 per cent from the assumed value of 2.605. The intermediate pressure will be $2.625 \times 14.73 = 38.7$ lb. per sq. in. abs., which is only approximately 0.3 lb. per sq. in. higher than the theoretical intercooler pressure computed from equation 30. For all practical purposes we may consider that the adoption of the $6\frac{1}{2}$-in. cylinder for the second stage of compression would be a satisfactory solution of the problem given above, especially since the area of the piston rod has not been considered.

Energy Relations. All two-stage energy computations must necessarily be based on polytropic conditions, as there would be very little reason for two-staging if the isothermal condition were obtainable.

Two-stage compression may be considered as two cylinders working separately on different ratios of compression. The energy required is thus the sum of the horsepower for the two cylinders added together. Transforming equation 16, for two-stage compression,

$$HP = \frac{0.0643\, n\, V_1}{n - 1} [R_1{}^{(n-1)/n} + R_2{}^{(n-1)/n} - 2] \tag{34}$$

in which V_1 is taken in cubic feet per minute and R_1 and R_2 are the compression ratios in the first and second stages. For the work to be equally divided between the stages, $R_1 = R_2 = \sqrt{R_t}$, where R_t is the overall ratio of compression. Assuming equal work in the two stages of compression, equation 34 becomes

$$HP = \frac{0.1286\, n\, V_1}{n - 1} [R_t{}^{(n-1)/2n} - 1] \tag{35}$$

in which V is expressed in cubic feet per minute.

When V is expressed in M.c.f.

$$HP = \frac{0.0892\, n\, V_1}{n - 1} [R_t{}^{(n-1)/2n} - 1] \tag{36}$$

Equations 34, 35, and 36 are based on double-acting compressors.

Displacements of the two cylinders in two-stage compression are figured separately. Capacity and overall volumetric efficiency are based solely on the low-pressure cylinder.

PROBLEMS

1. An air compressor intake is approximately 70°F., with a barometer of 28.5 in. of mercury. Assume that $n = 1.30$. It is desired to limit the discharge temperatures to 400°F. What is the highest discharge pressure that may be permitted? *Solution:* Temperature rise factor is $(400 + 460)/(70 + 460) = 1.624$. By looking down column marked "$n = 1.30$" in Table XI, we find a ratio of compression of approximately 8. Maximum discharge pressure would be $(8 \times 14) - 14 = 98$ lb. per sq. in. gauge. It would probably be safe to say the answer would be 100 lb. per sq. in., as the value 1.624 is slightly higher than the factor given in the table. Solving the proper equations for the exact value of R would not be justified in this problem.

2. A two-stage compressor takes in gas at 5 lb. per sq. in. gauge and discharges at 213 lb. per sq. in. gauge. What is the theoretical intercooler or intermediate pressure? *Solution:* Overall ratio of compression is $(213 + 14.6)/(5 + 14.6) =$

11.6. Intercooler pressure is $(5 + 14.6) \times \sqrt{11.6} = 66.8$ lb. per sq. in. abs. = 52.2 lb. per sq. in. gauge. The same result may be obtained by taking the square root of the product of the initial and final pressures. Intercooler pressure is $\sqrt{227.6 \times 19.6} = 66.8$ lb. abs. or 52.2 lb. gauge.

3. The ratio of compression for two-stage work is assumed to be 14. If the low-pressure cylinder is to be 12 in. in diameter, what will be the approximate size of the high-pressure cylinder? *Ans.:* 6.2 in. in diameter.

4. What is the theoretical horsepower required to compress 600 standard cubic feet per minute of dry natural gas from 20 to 260 lb. per sq. in. abs., assuming two-stage compression with an exponent of compression of $n = 1.2$? *Solution:* The ratio of compression for each stage is $\sqrt{260/20} = 3.6$. To use the tables, we may consider two separate compressions of 3.6 ratio of compression each. Horsepower required is $2 \times 600 \times 0.0917 = 110$ (from Table IV).

5. What is the temperature rise when wet natural gas is compressed from 10 lb. abs. and 65°F. to 88 lb. per sq. in. abs., assuming two-stage compression and a value of $n = 1.1$? *Solution:* Assume perfect intercooling in both stages. Overall ratio of compression is $88/10 = 8.8$, and ratio for each stage is $\sqrt{8.8} = 2.8$. Using Table XI, final temperature of compression is $(460 + 65) \times 1.0956 = 575°$ abs. or 115°F.

6. What would be the final temperature in problem 5 if the compression were single-stage? *Ans.:* 180°F.

CHAPTER IX

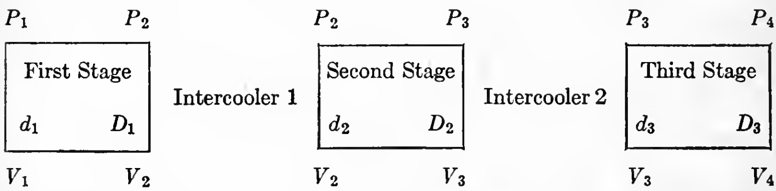
MULTISTAGE COMPRESSION

When the overall ratio of compression exceeds 10, it is often desirable to introduce another stage beyond two-stage compression, in order to cut down the temperature rise and save wear and tear on the machines.

Three-Stage Compression. For equal work in each stage of three-stage compression,

$$R_1 = R_2 = R_3 = \sqrt[3]{R_t} \tag{37}$$

where R_1, R_2, and R_3 are the ratios of compression in the three stages, and R_t is the overall compression ratio. This relation holds good only for equal clearance in all cylinders, and consequently equal volumetric efficiency.



Substitute in equation 37 the pressures corresponding to R_1, R_2, etc. We find that the pressure in the first intercooler is

$$P_2 = P_1\sqrt[3]{\frac{P_4}{P_1}}$$

$$P_2 = P_1\sqrt[3]{R_t} \tag{38}$$

By similar means, we find that the pressure in the second intercooler is

$$P_3 = P_1\sqrt[3]{R_t^2} = P_1 \times R_t^{2/3} \tag{39}$$

The temperature rise in three-stage compression is

$$T_2 = T_1 R_t^{(n-1)/3n} \tag{40}$$

where T_1 and T_2 are the initial and final temperatures (absolute) in each stage.

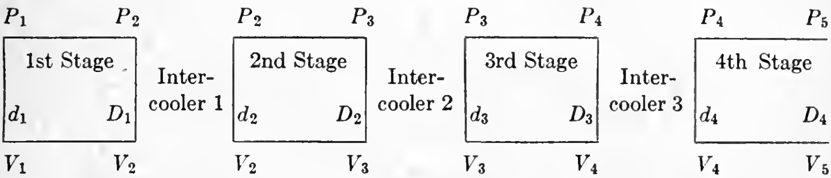
The power required in three-stage compression is

$$HP = \frac{0.1929\ n\ V_1}{n - 1} [R_t^{(n-1)/3n} - 1] \tag{41}$$

where V is taken in cubic feet per minute.

$$HP = \frac{0.1338\ n\ V_1}{n - 1} [R_t^{(n-1)/3n} - 1] \tag{42}$$

where V is taken in thousands of standard cubic feet per 24 hours (M.c.f.).



Four-Stage Compression. For extremely high ratios of compression, four stages are often found desirable. For equal work in each stage,

$$R_1 = R_2 = R_3 = R_4 = \sqrt[4]{R_t} \tag{43}$$

By substituting pressures for ratios of compression in equation 43, the pressure in the first intercooler is found to be

$$P_2 = \sqrt[4]{P_1{}^3 P_5} = P_1 \sqrt[4]{R_t} \tag{44}$$

The pressure in the second intercooler is

$$P_3 = \sqrt{P_1 P_5} = P_1 \sqrt{R_t} \tag{45}$$

The pressure in the third intercooler is

$$P_4 = \sqrt[4]{P_1 P_5{}^3} = P_1 R_t^{3/4} \tag{46}$$

The temperature rise for four-stage compression, with perfect intercooling between stages, is

$$T_2 = T_1\ R_t^{(n-1)/4n} \tag{47}$$

Energy relations for four-stage compression:

$$HP = \frac{0.2572\ n\ V_1}{n - 1} [R_t^{(n-1)/4n} - 1] \tag{48}$$

where V is taken in cubic feet per minute.

$$HP = \frac{0.1784\ n\ V_1}{n - 1} [R_t^{(n-1)/4n} - 1] \tag{49}$$

where V is taken in thousands of cubic feet per 24 hours.

Ideal Cylinder. All multistage compression may be considered as combinations of one- and two-stage problems, and treated accordingly. Equation 33 becomes hopelessly complicated when expanded to cover three- and four-stage compression.

The most direct method of determining cylinder sizes is to assume that the ratio of compression is the same in each cylinder. As the clearances of the different cylinders are known or may be determined, it is then possible to compute the volumetric efficiencies of each cylinder from equation 23.

Assuming that the strokes of each of the different cylinders are the same, the displacements vary as the square of the cylinder diameter, multiplied by the volumetric efficiency. After deciding upon the size of the low-pressure cylinder, it is convenient to compute the size of an ideal cylinder of equal capacity, but with 100 per cent volumetric efficiency. Then

$$d^2 E_v = d_e^2 \times 1.00$$

$$d_e = d\sqrt{E_v} \tag{50}$$

Using values obtained from equation 50, the sizes of other cylinders may be computed, assuming 100 per cent volumetric efficiency.

For three-stage compression,

$$d_2 = \frac{d_1}{R_t^{1/6}} \tag{51}$$

$$d_3 = \frac{d_2}{R_t^{1/6}} = \frac{d_1}{R_t^{1/3}} \tag{52}$$

For four-stage compression,

$$d_2 = \frac{d_1}{R_t^{1/8}} \tag{53}$$

$$d_3 = \frac{d_2}{R_t^{1/8}} = \frac{d_1}{R_t^{1/4}} \tag{54}$$

$$d_4 = \frac{d_3}{R_t^{1/8}} = \frac{d_1}{R_t^{3/8}} \tag{55}$$

The cylinder sizes as computed by equations 51 to 55, inclusive, are only " equivalent diameters," or the diameters of cylinders with 100 per cent volumetric efficiency. From equation 50, it is evident that we must divide these diameters by the square root of the corresponding volumetric efficiency, in order to get the actual cylinder to be used.

Piston Rod. In computing cylinder sizes, no attention has so far been paid to the size of the piston rod. In high-pressure cylinders,

the volume of the rod becomes increasingly important. It is evident that

$$\frac{2\pi d_c^2}{4} = \frac{2\pi d^2}{4} - \frac{\pi r^2}{4}$$

where d_c is the computed diameter, not considering the rod, and r is the rod diameter.

Then

$$d_c^2 = d^2 - \frac{r^2}{2}$$

$$d_c = \sqrt{d^2 - \frac{r^2}{2}}$$

$$d = \sqrt{d_c^2 + \frac{r^2}{2}} \qquad (56)$$

where d is the diameter of cylinder desired.

After computing cylinder sizes with the above equation, we find that the problem is further complicated by the fact that cylinders of the exact size required cannot as a rule be had, and that it is usually necessary to use manufacturer's sizes, which vary by even inches and half inches, except in the smaller sizes. From a similar problem in two-stage compression, it is evident that, in many cases, the deviation from calculated pressures will not be very great. At any rate, it is evident that the labor necessary to compute intercooler pressures and cylinder sizes for three- and four-stage compression to greater exactness than is attained by the foregoing formulas is certainly not justified.

PROBLEMS

1. A three-stage air compressor operating at about 250 ft. above sea level discharges at 330 lb. per sq. in. gauge. What pressures should we expect in the intercoolers, assuming equal work in each stage? *Solution:* Overall ratio of compression is $(330 + 14.6)/14.6 = 23.6$. Pressure in first intercooler is 27.2 lb. per sq. in. gauge; in second intercooler, 105.7 lb. per sq. in. gauge.

2. What horsepower is required to compress 650 standard cubic feet of air per minute from 14.6 lb. per sq. in. abs. to 400 lb. per sq. in. gauge, assuming three-stage compression and $n = 1.31$? *Ans.:* 160 hp. As a rough check, use Table IV for a single stage with compression ratio of 3 and value of $n = 1.30$, and multiply the result by 3.

3. The low-pressure cylinder for three-stage compression is 20 in. in diameter, with a computed volumetric efficiency of 90 per cent. The volumetric efficiencies of the second and third stages are 80 and 70 per cent, respectively. The overall ratio of compression is 30, and the piston rod is 3 in. in diameter. Required,

the sizes of the second- and third-stage cylinders. *Solution:* A 20-in. cylinder with 3-in. piston rod is equivalent to a hypothetical rodless cylinder 19.88 in. in diameter. Equivalent sizes for 100 per cent volumetric efficiency are 18.85-in., 10.68-in., and 6.05-in. Dividing by volumetric efficiencies to the one-half power, we get diameters of 19.88 in., 11.95 in., and 7.23 in., which represent "rodless" cylinders. Computed sizes with piston rods are 20-in., 12.17-in., and 7.53-in. Sizes used would probably be 20-in., 12-in., and $7\frac{1}{2}$-in.

4. What intercooler pressures are to be expected in a four-stage air compressor taking air at a barometer of 27.5 in. of mercury and discharging at 1000 lb. per sq. in. gauge, if the value of n is taken as 1.31? *Ans.:* 26.25, 103.5, and 331 lb. per sq. in. gauge.

5. What is the theoretical horsepower required for the compressor in problem 4, if its capacity is 1,000,000 standard cubic feet per day? *Ans.:* 219.5 hp. Check roughly by using $R = 3$ and $n = 1.30$ in Table V, and multiplying result by 4.

6. The compressor in problems 4 and 5 has a stroke of 20 in. and a low-pressure cylinder 15 in. in diameter, with 5 per cent clearance. Piston rod is $2\frac{1}{4}$ in. in diameter. At what speed does the machine operate? *Solution:* Volumetric efficiency of low-pressure cylinder is 93.6 per cent. Use equation 20 with added capacity correction of 27.5/30. Speed is 195 r.p.m.

7. What cylinder sizes are required in the second, third, and fourth stages of the compressor in problems 4, 5, and 6, if their respective clearances are 10, 15, and 20 per cent? *Solution:* A 15-in. cylinder with a $2\frac{1}{4}$-in. piston rod is equivalent to a hypothetical rodless cylinder 14.9 in. in diameter. Volumetric efficiencies are 93.6, 87.2, 80.8, and 74.4 per cent in the first, second, third, and fourth stages. The equivalent size of the low-pressure cylinder for 100 per cent volumetric efficiency is 14.4 in. Equivalent sizes for other cylinders, considering them "rodless," are 8.4-in., 4.9-in., and 2.855-in. Dividing by square roots of volumetric efficiencies, the sizes are 14.9-in., 8.95-in., 5.45-in., and 3.31-in. Allowing for rods, the sizes are 15-in., 9.08-in., 5.67-in., and 3.68-in. Actual sizes would probably be 15-in., 9-in., $5\frac{1}{2}$-in., and $3\frac{1}{2}$-in.

CHAPTER X

EXPONENT OF COMPRESSION

An examination of the equations in this book will show that numerical results in the expressions for brake horsepower, temperature rise, and volumetric efficiency all depend very largely upon the correct evaluation of the exponent n, which is commonly used in connection with the ratio of compression R. The value of this exponent cannot be set arbitrarily but depends upon a number of interrelated factors, such as the assumptions made in deriving the fundamental equations, the methods of allowing for deviation from the ideal gas laws, and the degree of cooling in the compressor.

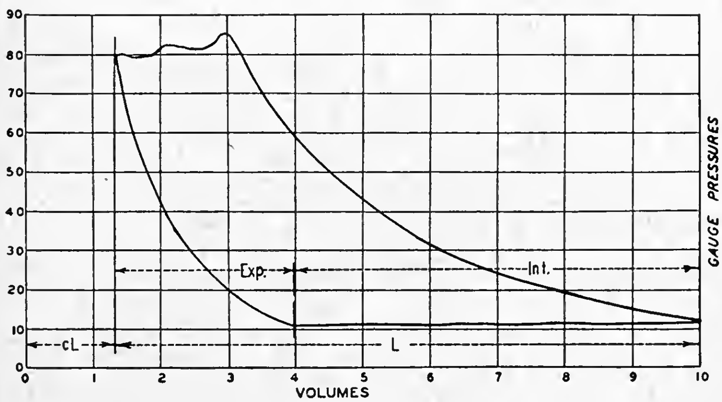


FIG. 15.

The indicator card, though not infallible, is probably as reliable a source of information of the operation of a compressor as we possess at the present time. Carefully taken cards should give valuable information about the numerical value of the exponent of compression. The indicator card does not tell the whole story, however. It should be considered in connection with all other data available.

At the outset, it will be well to state that it is usual to expect small discrepancies in values taken from indicator cards. Small variations

occur during seemingly constant load conditions. Consequently it will be advisable to take as many cards as possible and use the mean of

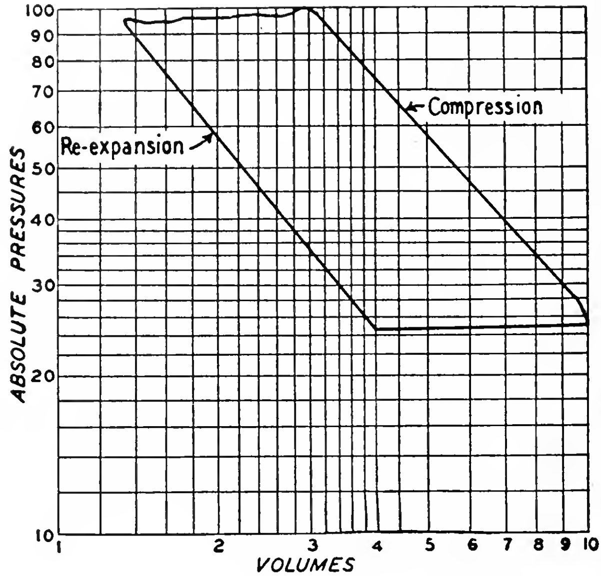


Fig. 16.

several readings. It is practically imperative to take both head and crank end cards.

After the indicator card is taken, the atmospheric line should be gone over with ink and a ruling pen for ease in plotting. Refer to Fig. 12. As we know the scale of the indicator spring, it is now possible to plot the line of zero pressure, absolute, which is drawn in parallel to the atmospheric line. If the clearance is known, we can find the line of zero volume, which will be perpendicular to the atmospheric line. The clearance is a percentage of the length of the stroke. If the length of the card L is 3.5 in. and the clearance is 8.5 per cent, then the line of zero volume is $cL = 3.5 \times 0.085 = 0.2975$ in. from the expansion end of the card, as projected upon the atmospheric line. After drawing in the line of zero volume, erect evenly spaced ordinate lines parallel to it, and it will then be possible to plot the indicator card on logarithmic cross-section paper. Figure 15 is an indicator card with gauge pressure and volume lines drawn in. Figure 16 is the same card, plotted on log-log paper. The slopes of the compression and expansion curves, as shown in Fig. 16, determine the value of n. For any curve on log paper, such as MN on Fig. 17 we may compute the slope as follows: Take any

straight portion of the curve AB, and disregard small curved portions at each end. Draw a line through A parallel to the OY axis, and another line through B parallel to the OX axis, the two lines intersecting at C. Measure the length of AC and BC in inches. The slope of AB is then $\overline{AC}/\overline{BC}$, or y/x. It will be noted that, if the angle ABC is 45°, the slope is 1.00; if the angle is greater than 45°, the slope is greater than 1.00.

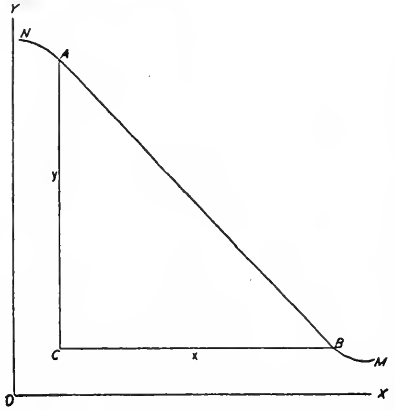


FIG. 17.

If logarithmic paper is not available, the value of n may be computed from the relation

$$n = \frac{\log P_b - \log P_a}{\log V_a - \log V_b} \qquad (57)$$

where pressures P and volumes V are taken at any points a and b along the curve to be analyzed.

It will often be found that the value of n does not plot a straight line on log paper, and consequently the value of n is not constant for the entire compression or expansion curve. This may be due to ring leakage or a number of other causes.

A frequent source of error is due to the fact that clearances are usually measured when the compressor is cold. When the machine is in operation, the piston rod may expand, increasing the clearance on one end and decreasing it on the other. In such cases, cards must be taken from both head and crank ends. The sum of the clearance volumes at both ends will remain the same, regardless of the expansion of the rod. If cards from head and crank ends show different exponents of compression, the values of clearance should be adjusted until both cards show the same slope when plotted on log paper. It is advisable to take the cards from head and crank ends at as near the same time as possible. Indicator mechanisms should preferably be actuated by a steel tape and should be installed in such a way as to eliminate parallax and angularity. Several manufacturers have provided places for indicator motions on their machines and can be relied upon to advise the best methods of operation.

If it is impossible to take a satisfactory number of indicator cards, and if no other data are available, reasonable results may be obtained from the relation

$$n = 0.75\ (k - 1) + 1 \qquad (58)$$

where n is the exponent of compression and k is the adiabatic value, or ratio of specific heats, usually given in the tables of properties of gases. For mixtures of different gases, as encountered in the natural-gas industry, it will be necessary to interpolate according to the specific gravity of the gas compressed. Equation 58 gives a value of approximately 1.31 for air.

It will often be found that values of n are different for the compression and re-expansion curves. The plotting of the re-expansion curve on log paper is usually difficult and often unreliable on account of pulsation and faulty valve action. In this connection, let us review the derivation of equation 15, which gives the theoretical energy required for polytropic compression.

$$W = \frac{n}{n-1} 2121\, V_1 \,[R^{(n-1)/n} - 1] \text{ ft-lb.} \tag{15}$$

Refer to Fig. 7. As the work done along the line AB includes the volume V_1 of the gas compressed as well as the gas in the clearance, cV_1, the actual work performed along this line AB is $(1 + c)\,W$. The negative work of re-expansion, which acts on the other side of a double-acting cylinder, would be $-cW$. The combination of the work of $(1 + c)\,W$ and $-cW$ gives W, which is expressed by equation 15. It is evident, however, that equation 15 does not hold good unless the exponent of compression n for the re-expansion CM is the same as that of the compression curve AB. The correct form of equation 15 would therefore be

$$W = (1 + c)\left\{\frac{a}{a-1} 2121\, V_1 \,[R^{(a-1)/a} - 1]\right\} - c\left\{\frac{b}{b-1} 2121\, V_1 \,[R^{(b-1)/b} - 1]\right\} \tag{59}$$

where a is the value of n for the compression curve, and b the value of n for the re-expansion curve. Equation 59 is too complicated to justify its use in the majority of cases, as the difference between a and b is usually not very large. However, it might be useful if values of n on compression and re-expansion should be found to vary considerably.

When we review the derivation of equation 23 for volumetric efficiency, it is found that we dealt entirely with the re-expansion curve. We will therefore be justified in using the value of n determined for re-expansion for computing volumetric efficiency.

CHAPTER XI

VOLUMETRIC EFFICIENCY AND COMPRESSOR BRAKE HORSEPOWER

The term "compressor brake horsepower" is taken as the energy that must be supplied by the prime mover, such as the gas-engine or electric motor, in order to run the compressor. Besides the power needed to compress the gas in the cylinder, the term includes valve losses and pipe friction in cylinder ports, as well as other miscellaneous mechanical losses in the compressor proper. The indicated horsepower, as shown by the indicator card, when divided by the mechanical efficiency, should give the brake horsepower of compression. It is evident that under the mechanical efficiency we must also include valve losses and pipe friction, which do not usually fall under this classification, but which must be included as power must be supplied to overcome them.

In the natural-gas industry, as well as elsewhere, it is customary to speak of "horsepower per million," or compressor brake horsepower required to compress 1 million standard cubic feet of gas per 24 hours within the pressure and temperature ranges desired.

A number of empirical equations have been proposed for compressor horsepower, among them being that of Thomas R. Weymouth,[1] published in 1912. It is now generally agreed that Weymouth's values are somewhat low, as his experiments were based on compressors having mechanically operated valves. The losses on the modern type of valves are somewhat higher than his equation would indicate.

A comparison of various data on the horsepower of compression[2] shows a rather wide variation between figures used by different compressor manufacturers and the Weymouth equation just mentioned, leading one to suspect that each curve is based on a separate series of experiments, which necessarily embody the peculiarities of each particular machine. We are therefore left the alternative of conducting experiments of our own, or of taking an average of published data, at the same time considering the problem in the light of what we know regarding the operation of the machine and the losses we are likely to encounter.

[1] *Transactions of the American Society of Mechanical Engineers*, vol. 34.

[2] See paper entitled "Compressors," read by Lyman F. Scheel at the October meeting of the California Natural Gasoline Association, held in Los Angeles, 1936.

Volumetric Efficiency. Tests on compressors operating with both intake and exhaust valves open to the atmosphere have indicated volumetric efficiencies averaging about 98 per cent, regardless of the gas compressed. This would lead us to modify equation 23 to read

$$E_v = 0.98 - c\,[R^{1/n} - 1] \tag{60}$$

where c is the clearance expressed as a decimal and R is the ratio of compression. Table XII gives values of E_v for an exponent of compression of $n = 1.20$, using equation 60.

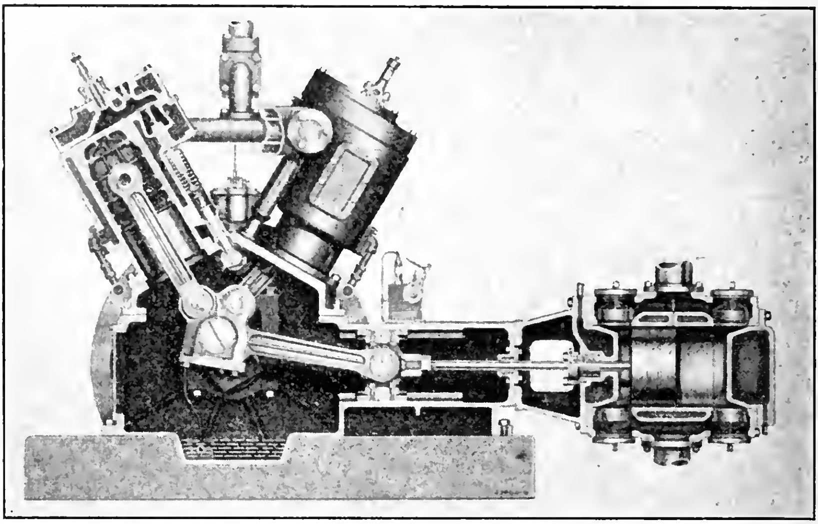


FIG. 18. Sectionized view of Ingersoll-Rand XVG gas-engine-driven compressor.

Actual volumetric efficiencies, determined from indicator cards and overall metering, are usually somewhat lower than values given by the theoretical equations, owing principally to ring leakage and valve losses. Some engineers seem inclined to use the isothermal ($n = 1.00$) for computing volumetric efficiencies, sometimes reducing the constant 0.98 in equation 60 as low as 0.90. Others use empirical equations which apparently have values of n less than 1.00.

The author has analyzed some recent volumetric efficiency tests, which gave empirical relations as follows:

Dry natural gas, specific gravity 0.75:

$$E_v = 0.97 - c\,[0.95\,R^{0.85} - 1]$$

Wet natural gas, specific gravity 0.84:

$$E_v = (1.05 - 0.044\,R) - c\,[1.18\,R^{0.637} - 1]$$

Air:

$$E_v = (0.95 - 0.0165\,R) - c\,[0.78\,R^{0.79} - 1]$$

This leads us to suggest a type equation for volumetric efficiency:

$$E_v = (1.00 - b - xR) - c\,[K\,R^{1/n} - 1] \tag{61}$$

where b, x, and K are constants which vary with the gas compressed and the operating characteristics of the individual compressor. The principal causes of the variation of these constants seem to be valve and intake manifold losses and ring leakage. The gas escaping past the piston rings into the other side of the cylinder often increases the apparent volumetric efficiency and also influences the horsepower per million cubic feet per 24 hours compressed, especially at higher ratios of compression.[1]

A method of deriving empirical equations from volumetric efficiency data is given in Fig. 19. Graph A shows the plotting of volumetric efficiencies, as a function of clearance, for three different ratios of compression. Ordinarily the observed points should plot as straight lines which intersect at some point X outside the graph and cross the line of zero clearance at points A, B, and C. The slope of these lines is given by the type formula $y = mx + b$, in which y is the volumetric efficiency, E_v; x is the clearance, c; b is the value of E_v where the line crosses the y axis; and m is the slope of the line, or $(K\,R^{1/n} - 1)$. Since $m = K\,R^{1/n} - 1$, then $K\,R^{1/n} = m + 1$.

Points A, B, and C give an indication of the combined ring and valve losses for the ratios of compression which they represent.

The next step is to plot points A, B, and C against ratios of compression, as in Graph B, and to determine the equation of the line through them. For the wet gas given above, this equation is $(1.05 - 0.044\,R)$. Graph C shows the plotting on log paper of one plus the slopes of the lines on Graph A for $R = 2$, $R = 4$, and $R = 6$. As given above, $m + 1 = K\,R^{1/n}$. These values are plotted against R. The slope of this line on log paper gives the exponent of R, and the point where the line crosses the line for $R = 1.00$ gives the value of K. In the wet gas equation, this plotting[2] gives the parenthesis $(1.18\,R^{0.637} - 1)$.

[1] See paper by Scheel just cited.

[2] In its present form, the equation for the wet gas fits the observed points closely and is therefore satisfactory as an empirical equation. The value of n is $1/0.637 = 1.57$, which is obviously too high. To get an idea of the real value of n, so as to check it with values obtained by other methods, we may make a slight adjustment of the point X in Fig. 19, to get 1.00 instead of 1.18 as the multiplier of R. This will change the entire system of constants in the equation but will give a much better value of n. The student is advised to try the replotting of this equation, the better to familiarize himself with the handling of empirical equations.

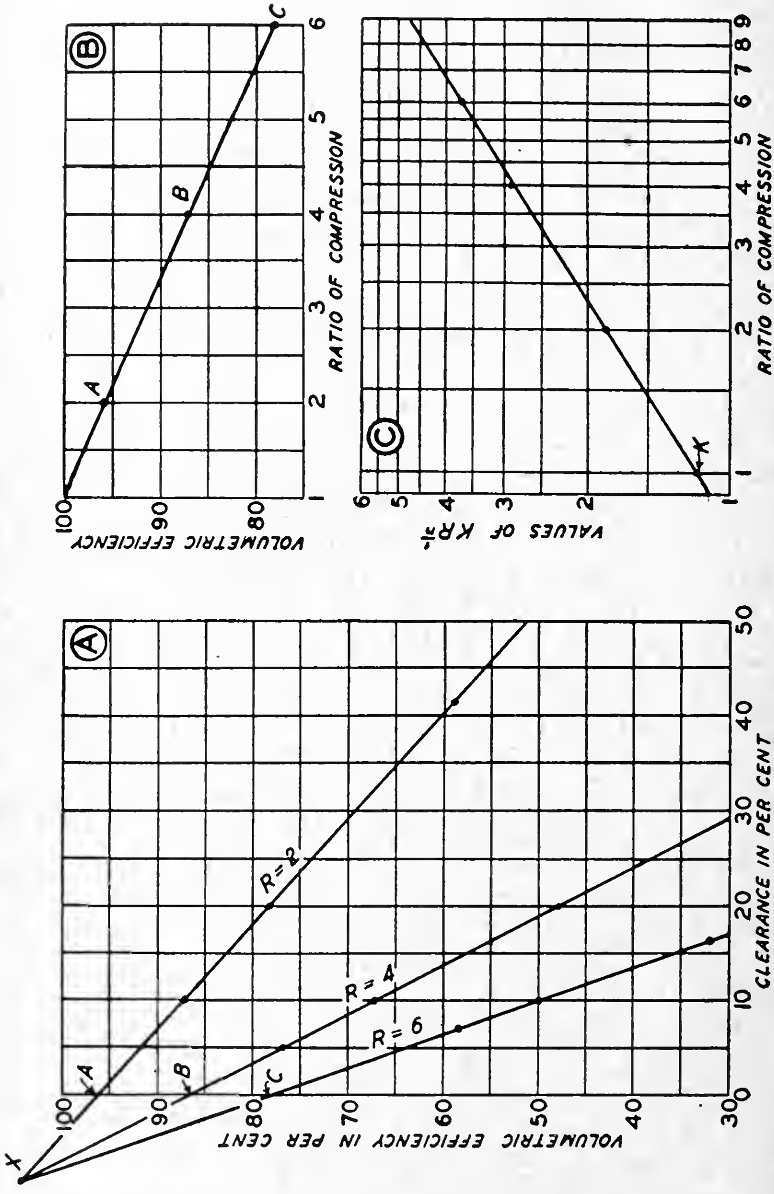

A
B
C
VOLUMETRIC EFFICIENCY
RATIO OF COMPRESSION
VALUES OF KR
RATIO OF COMPRESSION
R=2
R=4
R=6
VOLUMETRIC EFFICIENCY IN PER CENT
CLEARANCE IN PER CENT

FIG. 19.

Brake Horsepower. With atmospheric intake and exhaust, tests have shown that between 10 and 12 hp. per 1000 M.c.f. are usually required to overcome mechanical and valve and intake losses. Our curve for brake horsepower per million cubic feet per 24 hours must therefore begin at about 10 hp. for a ratio of compression of 1.00. It is reasonable to expect similar losses at other ratios of compression. Consequently we must add the same 10 hp. to theoretical values from equations 11, 12, 16, and 17. The 10 hp. is only a rough average and should be replaced by a more accurate value, if obtainable. As in volumetric efficiency equations, we must also introduce a term which is a function of the ratio of compression.

The following empirical equation will probably give a fair average for ordinary conditions:

$$BHP \text{ per } MMcf = \left\{\frac{44.6\,n}{n-1}\left[R^{(n-1)/n} - 1\right] + \text{Losses}\right\} f(R) \qquad (62)$$

where BHP per $MMcf$ is the brake horsepower of compression per million cubic feet per 24 hours,[1] and R is the ratio of compression. Values of brake horsepower per million, based on valve and mechanical losses of 10 hp. in equation 62, and with $f(R) = 1 + R^2/350$, are given in Table XIII. Equation 62 should not be used beyond a ratio of compression of 10.

Equation 62 takes no account of the piston speed of the compressor. Although variable-speed devices are now being introduced, it is customary to operate compressors within rather narrow limits of speed. Consequently the effect of piston speed must be considered one of the constants of the machine in evaluating valve and other losses, or in comparing one machine with another.

Very few data are at present available on valve and pipe friction losses during compression. There is no doubt that these losses vary widely in different machines, owing to differences of design. It is desirable, however, to limit the average velocity through the valves to approximately 100 ft. per sec. in order to keep valve losses from being excessive.

Other things being equal, we should look for smaller valve losses when the ratio of valve opening to cylinder cross section is high. It is the practice of many manufacturers to bore out several sizes of cylinders from the same casting, using the same valves for each. Cylinders in service are often bored out to a larger diameter after considerable use. Both these facts should be borne in mind when considering intake velocity and valve losses.

[1] For isothermal conditions, the equation becomes

$$BHP \text{ per } MMcf = (102.75 \log R + \text{Losses})\, f(R)$$

CHAPTER XII

COMPRESSOR PROBLEMS

Since extreme accuracy is difficult to obtain in compressor calculations, owing to lack of data on exact operating conditions and other causes, it has been found that many problems in compression may be satisfactorily solved with alignment charts, if sufficient care is taken to become familiar with their use. These charts are also very valuable in checking computations by other methods. For directions on use of charts, see Chapter XIX.

Chart 1. Ratio of Compression. This chart gives the value of R, the ratio of compression, or P_2/P_1.

PROBLEM 1. Intake of a compressor is 20 in. of mercury and the discharge is 20 lb. per sq. in. gauge. What is the ratio of compression? *Solution:* On Chart 1, use the two inside scales marked B. Draw a line from 20 in. on the left B scale to 20 lb. on the right B scale, and get a ratio of compression of 7.0 on the center scale.

Chart 2. Gas Volume Correction Chart. This chart is a graphical presentation of data in Table III, for reducing gas volumes to standard conditions.

PROBLEM 2. Reduce 187 cu. ft. of methane, at a pressure of 255 lb. per sq. in. abs., to standard cubic feet at 14.73 lb. per sq. in. abs. *Solution:* On Chart 2 draw a line from $(255 - 15) = 240$ lb. per sq. in. gauge on the left-hand scale to 1.87 on the right-hand scale, and get a volume of 32.5 on the center scale. As $187/1.87 = 100$, we must multiply the result by 100, giving us 3250 standard cubic feet. Compare this result with that of problem 1, Chapter III. See also Chapter XIII for deviation.

PROBLEM 3. A volumetric meter records 145 cu. ft. at 25 lb. per sq. in. gauge. How much gas does this represent in standard cubic feet? *Solution:* On Chart 2, draw a line from 25 on the left-hand scale to 1.45 on the right-hand scale and get 3.95 on the center scale. As $145/1.45 = 100$, multiply 3.95 by $100 = 395$ standard cubic feet.

PROBLEM 4. How much gas at 150 lb. gauge is represented by 1300 standard cubic feet? *Solution:* On Chart 2, draw a line from 150 on the left-hand scale through 13 on the middle scale, and get 1.15 on the right-hand scale. As 3001 is 100 times 13, multiply 1.15 by 100, making the result 115 cu. ft.

Chart 3. Theoretical Compression Horsepower. This chart is based on equations 11, 12, 16, and 17, and Tables IV and V.

PROBLEM 5. What is the theoretical isothermal horsepower required to compress 500 M.c.f. of natural gas from 14.7 to 73.5 lb. per sq. in. abs.? *Solution:* Ratio of compression is 73.5/14.7 = 5.0. On Chart 3, locate abscissa for ratio of compression of 5 on right side of chart. Follow this line upward from bottom until it intersects isothermal curve, and project at right angles to scale marked "function of (R,n)," obtaining a value of 0.0715. Draw a line from this value to 500 M.c.f. on left-hand scale, obtaining a theoretical horsepower of 35.5 on the middle scale. Compare with problem 2, Chapter IV.

PROBLEM 6. If it will theoretically require 40 hp. to compress 385 cu. ft. per min. of air from 14.6 to 58.4 lb. per sq. in. abs., what value of the exponent of of compression n is indicated? *Solution:* The ratio of compression is 58.4/14.6 = 4.0. On Chart 3, draw a line from 385 cu. ft. per min. on the left-hand scale, through 40 on the center scale, and get 0.073 on the "function of (R,n)" scale. Project this value to the right until it crosses the line for a ratio of compression of 4. The curve for n = 1.3 passes approximately through this point. The value of n is therefore 1.3.

PROBLEM 7. If an overall mechanical efficiency of 85 per cent is assumed, what would be the theoretical horsepower in problem 5? *Ans.:* 35.5/.85 = 41.8 hp.

Chart 4. Nominal Compressor Displacement. This chart is based on equations 19 and 20, with the term E omitted. A deduction of 2 per cent is made for the piston rod.

PROBLEM 8. What is the nominal displacement, in cubic feet per minute, of a compressor with 10-in. bore and 10-in. stroke, running at 275 r.p.m.? *Solution:* On Chart 4, draw a line from 10 on D to 10 on S, and find an intersection of 4.78 on A. From 4.78 on A draw a line to 275 on N, which crosses Q at 235 cu. ft. per min.

PROBLEM 9. If a compressor of 18-in. stroke runs at 180 r.p.m., what must be the size of the cylinder for 500 M.c.f. displacement? *Solution:* On Chart 4, draw a line from 180 on N to 500 M.c.f. on Q, and get an intersection of 5.92 on A. Draw a line from 18 in. on S through 5.92 on A, and get a cylinder diameter of approximately 11 in. on D.

Chart 5. Compressor Capacity and Displacement Chart. This chart is useful for converting capacity to displacement, or vice versa.

PROBLEM 10. The capacity of a compressor is 200 cu. ft. per min. The volumetric efficiency is 90 per cent, and the intake of the low-pressure cylinder is at 5 in. of mercury vacuum. What is the nominal displacement? *Solution:* On Chart 5, draw a line from 2 on the C scale to 90 on the E scale, and get an intersection of 2.32 on the turning scale A. From 5 in. vacuum on P draw a line through 2.32 on A, and get a value of 2.70 on D. As we used 2 instead of 200 on C, the answer will be 100 × 2.7 = 270 cu. ft. per min., nominal displacement.

PROBLEM 11. A compressor has a nominal displacement of 650 M.c.f. and a volumetric efficiency of 85 per cent. The intake to the low-pressure cylinder is 25 lb. per sq. in. gauge. What is the capacity of the machine? *Solution:* On Chart 5, draw a line from 25 lb. gauge on P to 6.5 on D, and get an intersection of 5.58 on A. Draw a line from 85 on E through 5.58 on A, and get a capacity of 15 on C. As we used 6.5 instead of 650 on D, the capacity is therefore $100 \times 15 = 1500$ M.c.f., or 1,500,000 standard cubic feet, per 24 hours.

PROBLEM 12. What is the capacity of a compressor of $12\frac{1}{2}$-in. bore and 18-in. stroke, running at 250 r.p.m. with volumetric efficiency of 85 per cent and intake at 47 lb. per sq. in. gauge? *Solution:* On Chart 4, draw a line from 12.5 on D to 18 on S, and get 6.3 on A. Draw a line from 6.3 on A to 250 on N, and get a displacement of 900 M.c.f. on Q. On Chart 5, draw a line from 9 on D to 47 on P, and get 6.75 on A. Draw a line from 85 on E through 6.75 on A, and get 32 on C. As we used 9 instead of 900 on D, the capacity is $100 \times 32 = 3200$ M.c.f. or 3,200,000 standard cubic feet per 24 hours. Compare result with problem 1, Chapter V.

PROBLEM 13. A compressor runs at 190 r.p.m. and has a 16-in. stroke. The volumetric efficiency is 88 per cent, and the intake is at 20 lb. per sq. in. abs. What cylinder size should be selected to compress 1,000,000 cu. ft. of gas per day (1000 M.c.f.) to 60 lb. gauge? *Solution:* On Chart 5, draw a line from 10 on C to 88 on E, and get 4.85 on A. Intake is $20 - 14.7 = 5.3$ lb. gauge. Draw a line from 5.3 on P through 4.85 on A, and get 8.1 on D. As we used 10 instead of 1000 on C, the nominal displacement is $100 \times 8.1 = 810$ M.c.f. On Chart 4, draw a line from 190 on N through 810 M.c.f. on Q, and get 6.53 on A. From 16 on S draw a line through 6.53 on A, and get a diameter of 14.5 in. on D. The alignment charts assume 2 per cent deducted for the volume of the piston rod. Compare results with problem 2, Chapter V.

Charts 6 and 7. Volumetric Efficiency. These charts solve for volumetric efficiency of gas compression, when the value of n and clearance are known. Chart 6 uses a value of $n = 1.16$ for dry gas and $n = 1.08$ for wet gas. Chart 7 has values of n from 1.0 to 1.5, and may be used for solving for n if desired. Both charts are based on equation 23.

PROBLEM 14. Dry natural gas is compressed from 10 in. vacuum to 30 lb. gauge in a compressor with 8.6 per cent clearance. What is the volumetric efficiency? *Solution:* The absolute discharge pressure is $30 + 14.7 = 44.7$ lb. Absolute intake pressure is $14.73 - (10 \times .491) = 9.82$ lb. per sq. in. abs. Ratio of compression is $44.7/9.8 = 4.55$. On Chart 6, draw a line from 4.55 on the left-hand side of the left-hand scale to 8.6 on the right-hand scale, and get a volumetric efficiency of 75.5 per cent on the center scale.

PROBLEM 15. Assume air compression, with $n = 1.30$, in problem 14. What is the volumetric efficiency? *Solution:* On Chart 7, draw a line from 4.55 on R through 1.30 on N, and get 2.25 on M. From 2.25 on A draw a line to 8.6 on C, and get a volumetric efficiency of 81 per cent on E.

PROBLEM 16. A compressor has a displacement of 800,000 cu. ft. per day at atmospheric intake. A meter on the intake line reads 700 M.c.f. It is assumed that there is no loss from shrinkage or condensation. The clearance is 8 per cent, and the ratio of compression is 3. What is the value of n? *Solution:* The volumetric efficiency is 700/800 = 87.5 per cent. On Chart 7, draw a line from 8 on C through 87.5 on E, and get 1.55 on A. Transferring this value to scale M, draw a line from 1.55 on M to 3 on R, and get 1.20 as the value of n.

NOTE. If the chart gives a value of n that falls below unity, then high valve and ring losses are indicated. See equation 61, Chapter XI.

PROBLEM 17. A compressor has a clearance of 8 per cent and a volumetric efficiency of 71.5 per cent. What clearance is necessary to reduce the volumetric efficiency to 50 per cent to prevent stalling of the machine? *Solution:* On Chart 7, draw a line from 8 on C through 71.5 on E, and get 3.55 on A. From 3.55 on A draw a line through 50 on E, and get a clearance of 14.5 per cent on C.

NOTE. As experience has shown that volumetric efficiency is more nearly given by equation 60 than by equation 23, it will be found that volumetric efficiencies computed by Charts 6 and 7 are high.

Chart 8. Maximum Ratio of Compression. This chart is based on equation 24.

PROBLEM 18. What is the maximum percentage of clearance for a ratio of compression of 16, assuming an air compressor operating at $n = 1.40$? *Solution:* On Chart 8, draw a line from 16 on the right-hand scale through 1.40 on the center scale, and get a maximum clearance of 16 per cent on the left-hand scale. See problem 2, Chapter VI.

PROBLEM 19. A compressor has a volumetric efficiency of 70.3 per cent and a clearance of 10 per cent. The exponent of compression n may be taken as 1.30 for air. The ratio of compression is 6. How many clearance pockets which increase the cylinder clearance by 5 per cent each may be opened into the cylinder before the compressor ceases to discharge air? *Solution:* On Chart 8, draw a line from 6 on the right-hand scale through 1.30 on the center scale, and get a maximum clearance of 33.7 per cent on the left-hand scale. Therefore four clearance pockets may be opened into the cylinder before discharge ceases. Compare problem 1, Chapter VI.

Chart 9. Temperature Rise of Gas During Compression. This chart is based on equation 29, assuming an exponent of compression $n = 1.26$ and an initial temperature of 80°F.

PROBLEM 20. What is the final temperature of gas compressed from atmosphere to 100 lb. per sq. in. gauge, assuming an initial temperature of 80°? *Solution:* On Chart 9, draw a line from atmosphere on the right-hand scale to 100 lb. gauge on the left-hand scale, and get a final temperature of 362°F. on the center scale.

Chart 10. Intercooler Pressure for Two-Stage Compression. Based on equation 30.

Problem 21. A two-stage compressor takes in gas at 5 lb. per sq. in. gauge and discharges at 213 lb. per sq. in. gauge. What is the theoretical intercooler or intermediate pressure for equal work in both cylinders? *Solution:* On Chart 10, draw a line from 5 lb. on the right-hand scale to 213 on the left-hand scale, and get a receiver pressure of 52.5 lb. per sq. in. gauge. Compare with problem 2, Chapter VIII.

Chart 11. Cylinder Sizes for Two-Stage Compression. Based on equation 32.

Problem 22. The overall ratio of a two-stage compressor is 14, and the low-pressure cylinder is 12 in. in diameter. What is the size of the high-pressure cylinder? *Solution:* On Chart 11, draw a line from 14 on the right-hand scale through 12 on the center scale and get 6.2 in. on the left-hand scale as the diameter of the high-pressure cylinder. Compare problem 3, Chapter VIII.

Chart 12. Volumetric Efficiency Correction for Compressor Cylinder Size. Based on equation 50. This chart computes the size of an "equivalent" cylinder of 100 per cent volumetric efficiency which has the same capacity as a known cylinder of known volumetric efficiency.

Problem 23. A 9.6-in. cylinder has a volumetric efficiency of 81 per cent under stated conditions of clearance and ratio of compression. What would be the equivalent size of a cylinder with the same capacity, but with 100 per cent volumetric efficiency? *Solution:* On Chart 12, go from 9.6 on the left-hand scale to 81 on the right-hand scale, and get 4.9 on the central turning scale. Draw a line from 100 on the right-hand scale through 4.9 on the center scale, and get 8.7 in. as the diameter of the equivalent cylinder.

Chart 13. Intercooler Pressures for Three-Stage Compression. Based on equations 38 and 39.

Problem 24. A three-stage air compressor discharges at 330 lb. per sq. in. gauge. What pressures should we expect in the intercoolers, assuming equal work in each stage? *Solution:* On Chart 13, draw a line from atmosphere on the left-hand scale to 330 on the right-hand scale, and get intercooler pressures of 27.5 and 105 lb. per sq. in. gauge. Compare with problem 1, Chapter IX.

Chart 14. Cylinder Sizes for Three-Stage Compression. Based on equations 51 and 52.

Problem 25. The low-pressure cylinder for three-stage compression is 20 in. in diameter, with a computed volumetric efficiency of 90 per cent. The volumetric efficiencies of the second and third stages are 80 and 70 per cent, respectively. The overall ratio of compression is 30. Required, cylinder sizes in the second and third stages. *Solution:* Disregard volumetric efficiencies.

On Chart 14, draw a line from 30 on the right-hand scale through 20 on the low-pressure cylinder scale, and get intermediate and high-pressure cylinder diameters of 11.4 and 6.5 in., respectively. Compare with problem 3, Chapter IX. Evidently the sizes are too small, so that it will be necessary to take the volumetric efficiency into account. Use Chart 12. Draw a line from 20 on the left-hand scale to 90 on the right-hand scale, and get 6.98 on the center scale. Draw a line from 100 on the right-hand scale through 6.98 on the center scale, and get an equivalent cylinder of 19.0 in. diameter on the right-hand scale. Now on Chart 14, draw a line from 30 on the right-hand scale through 19 on the low-pressure cylinder scale, and get an intermediate cylinder of 10.8-in. diameter and a high-pressure cylinder of 6.1-in. diameter. These are "equivalent" cylinders of 100 per cent volumetric efficiency. Use Chart 12. Draw a line from 100 on the right-hand scale to 10.8 on the left-hand scale, and get 5.46 on the center scale. Draw a line from 80 on the right-hand scale through 5.46 on the center scale, and get an actual intermediate cylinder of 12 in. diameter on the right-hand scale. Draw a line from 100 on the right-hand scale to 6.1 on the left-hand scale, and get 4.0 on the center scale. Draw a line from 70 on the right-hand scale through 4.0 on the center scale, and get 7.35 as the size of the high-pressure cylinder, from the left-hand scale. Comparison with problem 3, Chapter IX, will show that it might be desirable to increase the size of the high-pressure cylinder slightly from 7.35 in. in diameter to allow for the piston rod.

Chart 15. Intercooler Pressures for Four-Stage Compression. Based on equations 44, 45, and 46.

PROBLEM 26. What intercooler pressures are to be expected in a four-stage air compressor taking its intake at $2\frac{1}{2}$ in. of vacuum and discharging at 1000 lb. per sq. in. gauge? *Solution:* On Chart 15, draw a line from 2.5 in. of mercury vacuum on the left-hand scale to 1000 lb. gauge on the right-hand scale, and read off intermediate pressures of 25.3, 103, and 330 lb. per sq. in. gauge, respectively, for the first, second, and third intercoolers. Compare with problem 4, Chapter IX.

Chart 16. Cylinder Sizes for Four-Stage Compression. Based on equations 53, 54, and 55.

PROBLEM 27. An air compressor with an overall ratio of compression of 75 for four-stage compression has a 15-in. low-pressure cylinder. The volumetric efficiencies are: 93.6 per cent, low pressure; 87.2 per cent for second stage; 80.8 per cent for the third stage; and 74.4 per cent for the fourth or high-pressure stage. What size cylinders should be used in the second, third, and fourth stages? *Solution:* Judging from problem 25 for three-stage compression, we can make rather large errors if we disregard volumetric efficiencies, unless, of course, only approximate sizes are required. Using Chart 12, draw a line from 93.6 on the right-hand scale to 15 on the left-hand scale, and get 6.3 on the center scale. Draw a line from 100 on the right-hand scale through 6.3

on the center scale, and get an equivalent diameter of 14.4 in. on the left-hand scale. On Chart 16, draw a line from 75 on the right-hand scale, through 14.4 on the low-pressure cylinder scale, and get 8.4 in., 4.86 in., and 2.85 in. for the equivalent sizes of the three cylinders.

Now use Chart 12 to reduce these equivalent sizes to actual sizes. Draw a line from 100 on the right-hand scale to 8.4 on the left-hand scale, and get 4.82 on the center scale. From 87.2 on the right-hand scale, draw a line through 4.82 on the center scale, and get 9 in. on the left-hand scale as the actual size of the second-stage cylinder. Draw a line from 100 on the right-hand scale to 4.86 on the left-hand scale and get 3.4 on the center scale. From 80.8 on the right-hand scale draw a line through 3.4 on the center scale, and get 5.45 in. for the size of the third-stage cylinder. Draw a line from 100 on the right-hand scale to 2.85 on the left-hand scale, and get 1.95 on the center scale. Draw a line from 74.4 on the right-hand scale through 1.95 on the center scale, and get 3.3 in. on the left-hand scale as the size of the high-pressure cylinder. Comparing problem 7, Chapter IX, it will be seen that it is desirable to increase the size of the high-pressure cylinder somewhat to allow for the piston rod.

Chart 17. This chart is based upon a formula which contains all the important variables encountered in compression, and, though intended primarily for clearance problems, it may be used for other calculations as well. The brake horsepower of compression, for any compressor, may be found by multiplying the horsepower per million cubic feet from equation 62 by $Q/1000$. For Q, the capacity, we may substitute from equation 20,

$$\frac{Q}{1000} = \frac{0.001309\, d^2\, L\, N\, E_v\, P_1}{1000 \times 14.73}$$

where E_v is taken from equation 60. The complete equation becomes

$$HP = HP \text{ per } MMcf \times \frac{0.001309\, d^2\, L\, N\, E_v\, P_1}{1000 \times 14.73} \tag{63}$$

The alignment chart is based on an exponent of compression of $n = 1.2$.

PROBLEM 28. The clearance of a compressor is 9.5 per cent. It is equipped on each end of the cylinder with clearance chambers which increase the clearance by 2 per cent each when opened. The intake pressure is 80 lb. per sq. in. gauge, and the discharge is 350 lb. gauge. It is now operating at approximately 75 brake horsepower with all the clearance chambers closed. The maximum power available through the prime mover is 85 brake horsepower. How many clearance chambers must be opened to keep the machine from stalling from lack of power? *Solution:* On Chart 17, draw a line from 80 on N through 350 on O, and get a ratio of compression of 3.8 on R. On the graph on the right side of the chart, follow the R line for 3.8 up from the bottom until it crosses the clearance curves at 9.5 per cent, and project to the left, getting a value of 63 on the M

scale. From 63 on M draw a line to 80 on N, and get 5.9 on the turning scale A. From 5.9 on A draw a line through 75 on H and get an equivalent displacement of 185 M.c.f. on K. This value of 185 M.c.f. may not agree with the value of $d^2LN/1000$ for the machine in question but will serve for the problem at hand.

Now draw a line from 150 on N through 350 on O, and get a new ratio of compression of 2.2 on R. As we are limited to 85 hp., draw a line from 185 M.c.f. on K through 85 on H, and get 6.13 on the turning scale A. Draw a line from the new intake of 150 lb. on N through 6.13 on A, and get 40 on M. Follow the 40 line to the right until it intersects the 2.2 ratio of compression line at a clearance of about 16 per cent. The increase in clearance is $16 - 9.5 = 6.5$ per cent. We must therefore open four clearance chambers of 2 per cent each on each end to keep the machine from stalling. With the original clearance and all chambers closed, about 90 hp. would be required to run the machine.

On account of the unusually large number of variables on Chart 17, the accuracy cannot be very great. However, it serves to give an approximate solution to problems like 28, which are very tedious when solved by formula.

PROBLEM 29. Assuming that $n = 1.2$, what are the compressor brake horsepower and capacity of a machine with 20-in. stroke and 14-in. cylinder with 12 per cent clearance running at 204 r.p.m., with intake at 30 lb. per sq. in. gauge and discharge at 100 lb. gauge? *Solution:* First get the ratio of compression by drawing a line on Chart 17 from 30 on N through 100 on O, getting 2.55 on R. Follow upward along the 2.55 ratio of compression line to the intersection of the 12 per cent clearance line, and project to the left, getting 47 on M. From 30 on N draw a line to 47 on M, and intersect 4.1 on the turning scale A. Substitute in $d^2LN/1000$, getting a value of 800, which corresponds to a nominal displacement of 1050 M.c.f., as values on either side of scale K are interconvertible. From 1050 M.c.f. on K draw a line to 4.1 on the turning scale A and get 150 hp. on H. Refer to Table III, and get a pressure multiplier of 3.027 for an intake of 30 lb. gauge. From Table XII, for a clearance of 12 per cent, the volumetric efficiency is 83.8 per cent. The capacity is therefore $1050 \times 3.027 \times 0.838 = 2670$ M.c.f.

PROBLEM 30. A compressor takes suction at 10 in. of vacuum and discharges at 54 lb. gauge. The cylinder is 13 in. in diameter with 16.5 per cent clearance. The stroke is 16 in., and the machine runs at 252 r.p.m. What are the brake horsepower and capacity? What size prime mover will be required? *Solution:* On Chart 17, draw a line from 10 in. of vacuum on N through 54 on O, and get a ratio of compression of 7.0 on R. Follow the 7 ratio of compression line upward on the graph at the right side of the chart, until it intersects the line of 16.5 per cent clearance, and project to the left, obtaining 40 on M. Draw a line from 40 on M to 10 in. vacuum on N, and get 1.2 on the turning scale A. Substitute in $d^2LN/1000$, and get a value of 680, which corresponds to a displacement of 900 M.c.f. on K. From 1.2 on A draw a line to 900 M.c.f. on K, and get 25 hp. on H. This is the power required to run the machine at the conditions stated in the problem.

However, an examination of the curve for 16.5 per cent clearance shows that, as the pressure is being built up and the ratio of compression is being increased

in starting up the machine, a maximum value of the function of horsepower and volumetric efficiency is reached at a ratio of compression of about 4.0. The prime mover must therefore be able to supply power enough to get over this hump before the desired operating conditions can be reached. This maximum point when projected on M gives a value of 52. Solving for power from this point, we get a maximum of 30 hp. which must be furnished in order to reach steady operating conditions. If the prime mover will not supply this much power, we may keep within our 25 hp. by increasing the clearance and " unloading " the machine while starting up. By following the horizontal line for $M = 40$, as obtained above for 25 hp. at 16.5 per cent clearance, we find that a clearance of 25 per cent will reach its maximum just below the 40 line. In order to keep down the size of the prime mover, we could then provide clearance pockets to increase the clearance to 25 per cent and use them in starting the machine. The pockets should then be closed, unless it is desired to reduce the capacity of the machine for some other reason.

To compute the capacity, from Table III get a factor of 0.658 for a 10-in. vacuum intake. The volumetric efficiency at 16.5 per cent clearance and a ratio of compression of 7, from Table XII, is 29.3 per cent. The capacity is therefore $900 \times 0.658 \times 0.293 = 173.7$ M.c.f.

The above problem is typical of vacuum pumps, which ordinarily have a high ratio of compression and operate beyond the maximum of the horsepower-volumetric efficiency function, and consequently on the right side and down slope of the clearance curves in Chart 17. Problems concerning pumps working at more than 10 in. vacuum would run off the scales of Chart 17. The graph on the right side of this chart, however, would serve to locate the maximum power point, if the ratio of compression were not too high.

PROBLEM 31. An air compressor of 16-in. stroke and 12-in. bore delivers air at 59 lb. per sq. in. The intake is assumed to be atmosphere, and the clearance is 15 per cent. It is desired to run the compressor with a constant-speed motor, with clearance pockets arranged to reduce the capacity in 10 per cent steps by unloading. What size pockets are required? *Solution:* The ratio of compression is $(59 + 14.73)/14.73 = 5$. For an approximate solution, use Table XII, which calls for $n = 1.2$. The volumetric efficiency at $R = 5$ and a 15 per cent clearance is 55.6 per cent. The clearance chambers would be found from the table by successively subtracting 10 per cent of $55.6 = 5.56$ from the original volumetric efficiency and from the values of E_v later computed, and noting the corresponding values of clearance. Thus $55.6 - 5.56 =$ approximately 50, which corresponds to a clearance of 17 per cent. The next step is $50 - 5.56 = 44.44$, which is close enough to the tabular value of 44.3 at 19 per cent clearance. For the first two pockets, therefore, we will have 2 per cent of the cylinder volume, or

$$\frac{0.02\ \pi\ 12^2 \times 16}{4} = 36.2 \text{ cu. in.}$$

Using Table VIII to solve equation 60 with $n = 1.3$, we get a volumetric efficiency of 61.3 per cent at 15 per cent clearance. Taking 10 per cent from 61.3, we have 55.2 per cent. Subtract 0.552 from 0.98 and get 0.428, which, when divided by the tabular value for $n = 1.3$ from Table VIII, gives a clearance of approximately 17.5 per cent. This would mean a clearance pocket of $2.5/2 \times 36 = 45.2$ cu. in.

CHAPTER XIII

SUPERCOMPRESSIBILITY OR DEVIATION FROM THE IDEAL GAS LAWS

When gas is compressed much beyond 50 lb. per sq. in. gauge, there is usually a difference between the actual volume and the volume we should expect by applying equations 3 and 4, or the ideal gas law, $P V = wR T$ (equation 5). The difference is usually a shrinkage, accompanied by a corresponding increase in density. This shrinkage, or deviation, seems to be a function of both temperature and pressure, and reaches a maximum at the critical temperature and pressure of the gas.

Under certain conditions, some gases seem to resist compression and show larger volumes than would be expected. When air is compressed, we encounter an increase in volume above about 120°F. and a shrinkage below this temperature. The deviation of air is very small when compared with that of nearly all other gases.

The published data on supercompressibility of gases, which are rapidly being increased at the present time, are usually found under the three following classifications:

1. Supercompressibility of individual gases, such as given in publications of the Bureau of Standards.[1]
2. Orifice meter data,[2] which are concerned with the change of volume on actual or hypothetical expansion of the gas from stated conditions to the standard pressure and temperature of gas measurement. In gas measurement, the change in density is spoken of as superexpansibility.
3. Supercompressibility of gases or mixtures of gases as a function of the temperature in relation to the critical temperature, and of the pressure in relation to the critical pressure.

Supercompressibility, or volume deviation from the ideal gas law, may be expressed as δ and taken as a decimal, the value of which is

[1] "Compressibilities of Gases," *Misc. Publ.* Bureau of Standards, 71, 1925. "Relations between Temperature, Pressure and Density of Gases," *Circ.* Bureau of Standards, 279, 1926.

[2] "Tentative Standards for Determination of Superexpansibility Factors in High Pressure Gas Measurement," *Bull.* TS-354, California Natural Gasoline Association, Los Angeles, 1936.

usually less than unity for temperatures encountered in compression work. In helium, hydrogen, air, nitrogen, and several other gases at high temperatures, the deviation factor is greater than 1.00.

In gas-measurement work, the increase in volume on expansion may be taken as Δ, the total volume being $1 + \Delta$, consequently

$$\delta = \frac{1}{1 + \Delta} \tag{64}$$

The total effect of deviation is so closely connected with temperature that it is not convenient to consider the pressure-volume relationship separately.

In the deviation data of the Bureau of Standards, the abscissas represent pressures in atmospheres, the curves are isothermals, and the ordinates give the deviation factor, which is

$$\delta = \frac{273\,P\,V}{T} \tag{65}$$

where T is in degrees Centigrade. The base temperature is therefore 0°C. or 32°F. Since P is expressed in atmospheres, it may be replaced by P/P_s, where P is any pressure in pounds per square inch absolute, and P_s is the standard pressure of gas measurement. The 273°C. may be replaced by the standard temperature T_s, if T is taken in degrees Fahrenheit. The equation then becomes

$$\delta = \frac{V\,P\,T_s}{P_s\,T} \tag{66}$$

or, in terms of volume,

$$V = \frac{\delta\,P_s\,T}{P\,T_s} \tag{67}$$

Writing equation 67 twice, with subscripts 1 and 2 for all the variables except T_s and P_s, and dividing the first equation by the second,

$$\frac{V_1}{V_2} = \frac{\delta_1\,T_1\,P_2}{\delta_2\,T_2\,P_1} \tag{68}$$

in which T_s and P_s cancel out. The ratio P_2/P_1 is recognized as the ratio of compression, R. For convenience, we may let

$$Z = \frac{\delta_1\,T_1}{\delta_2\,T_2} \tag{69}$$

Then $V_1/V_2 = ZR$. From equation 7, $(V_1/V_2)^n = R$. Consequently,

$$\frac{V_1}{V_2} = R^{1/n}$$

Equating to eliminate V_1/V_2, $R^{1/n} = ZR$, which may be written

$$\frac{1}{Z} = R^{(n-1)/n} \tag{70}$$

Solving for n,

$$n = \frac{\log R}{\log ZR} \tag{71}$$

From equation 71 we may find the value of n if we know the initial and final temperatures of compression and the corresponding deviations.

Let us take for example a compressor operating on a natural gas of 0.90 specific gravity, with intake at 80°F. and 100 lb. gauge, and discharge at 150° and 300 lb. gauge. Deviation factors are 0.9690 for intake and 0.9366 for discharge. The value of R is 314.73/114.73 = 2.74. Substituting in equation 69,

$$Z = \frac{0.9690 \times 540}{0.9366 \times 610} = 0.917$$

Substituting in equation 71,

$$n = \frac{\log 2.74}{\log (2.74 \times 0.917)} = 1.095$$

To find the effect of deviation, let us solve for a value of the exponent of compression, to be known as x, which will give the same temperature rise as the value of n just determined, assuming no deviation. From equation 71, taking $\delta_1 = \delta_2 = 1.00$,

$$x = \frac{\log R}{\log R + \log T_1 - \log T_2}$$

Substituting values from the above example,

$$x = \frac{\log 2.74}{\log 2.74 + \log 540 - \log 610} = 1.138$$

Substituting in equation 29, with $R = 2.74$ and $n = 1.138$, we find $T_2/T_1 = 1.13$. For $T_1 = 80°F.$, $T_2 = 1.13 \times 540 = 610°$ abs., or 150°F., which is the same temperature drop as we found in the above example where deviation was present.

Thus we may conclude that:

1. For the same rise in temperature during compression, the exponent of compression n varies with the deviation factor, a high n indicating a factor above 1.00, and a low value indicating a factor below 1.00.

2. For the same value of n shown on an indicator card, a high temperature rise shows a deviation factor above 1.00, and a low rise shows a deviation factor below 1.00.

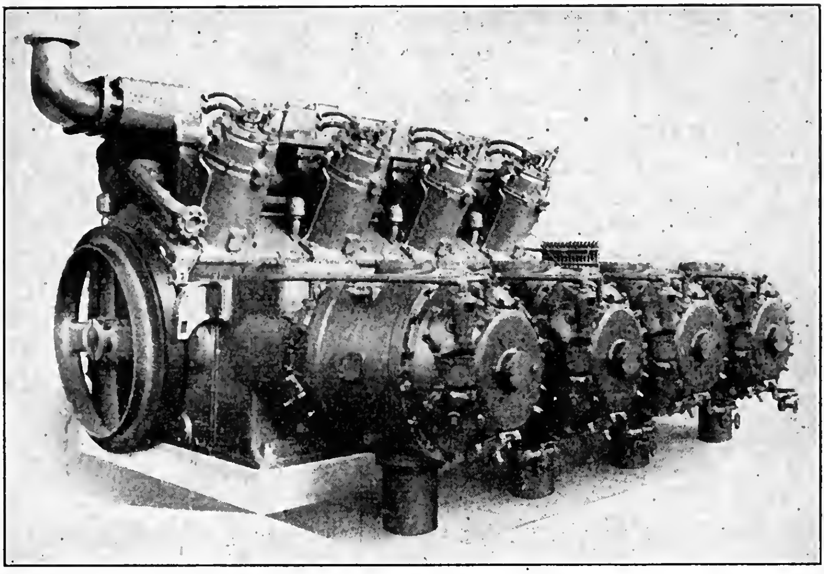


FIG. 20. Cooper-Bessemer type G-MV gas-engine-driven compressor unit — rated at 800 hp. at 300 r.p.m.

If we know the value of the exponent of compression n for a gas having deviation, and have a table of deviation factors, we can find the final temperature of compression, if the initial temperature is known. Writing Z (equation 69) in terms of deviation and temperature, we obtain from equation 70:

$$\delta_2 T_2 = \delta_1 T_1 R^{(n-1)/n} \tag{72}$$

As all variables on the right-hand side of equation 72 are known, the equation can be solved for $\delta_2 T_2$, by plotting a curve with ordinates $\delta_2 T_2$ and abscissas T_2. Such a curve will give a value of T_2 corresponding to the value of $\delta_2 T_2$ obtained from equation 72.

Deviation from Ideal Gas Laws. Since the ideal gas law (equation 5) was used in the derivation of nearly all the equations encountered in the theory of compression, it is evident that we must re-examine these equations to determine the effect of compressibility, or deviation from the ideal gas laws.

The first equation to be affected by deviation is the so-called perfect gas law itself, given in equation 5. Since deviation data from different

sources do not agree upon the temperature and pressure at which $\delta = 1.00$, it will be necessary to use the ratio of the deviation at the desired points with the deviation from the same data at standard conditions of 14.73 lb. per sq. in. and 60°F. This can be done by rewriting equation 68, obtaining

$$\frac{V}{V_s} = \frac{\delta\, T\, P_s}{\delta_s\, T_s\, P} \tag{73}$$

in which the subscripts s refer to standard conditions. In terms of standard volume,

$$V_s = \frac{V\,\delta_s\, T_s\, P}{\delta\, T\, P_s}$$

$$V_s = \left(\frac{520\,\delta_s}{14.73}\right)\left(\frac{P\,V}{\delta\, T}\right) \tag{74}$$

Usually δ_s will be 1.00. Equation 74 is valid for any weight of gas, as all expressions containing weight were cancelled out in the derivation of equation 68.

Compare problem 1, Chapter III, in which it is required to reduce 187 cu. ft. of methane, at a pressure of 255 lb. per sq. in. abs., to standard cubic feet at 14.73 lb. per sq. in. abs. The deviation of methane, from Table XV, is 0.968 for 255 lb. abs. (240 lb. gauge). We will consider the temperature as 60° in the initial condition of the gas. Substituting in equation 74,

$$V_s = \frac{520 \times 1.00 \times 255 \times 187}{14.73 \times 0.968 \times 520} = 3345 \text{ standard cubic feet}$$

Solving this problem without considering deviation, the final volume is 3237 standard cubic feet.

Intake Correction. The principal effect of deviation is seen in the term P_1V_1, which is equal to the constant K in equation 8. This term P_1V_1 was not included in the integration. V_1 represents the intake volume of gas in standard cubic feet when P_1 is taken as 14.73 lb. per sq. in. abs. The deviation between the term PV for the pressure and volume at any point, and P_sV_s the standard pressure 14.73 and the volume in standard cubic feet, may be expressed, by rewriting equation 69 as

$$Y = \frac{\delta_1\, T_1}{\delta_s\, T_s} \tag{75}$$

in which T_s and δ_s are the standard temperature of gas measurement and the deviation factor at this temperature (usually 1.00), and T_1 and δ_1 are the temperature and deviation at compressor intake.

Volumetric Efficiency. In analyzing an indicator card, it is customary to take the volumetric efficiency as the percentage of the compressor stroke during which the intake occurs. But, in deriving equation 23, we assumed that the volumetric efficiency was the ratio of the actual intake volume to the displacement, and, as it is universally so regarded, there would be considerable confusion if we should introduce an expression for deviation into the volumetric efficiency equation. It is more convenient to put the deviation into the capacity equation.

Compressor Capacity. Rewriting equation 21, we have

$$C = \frac{D\,E_v\,P}{14.73} \tag{76}$$

where C is the intake capacity in standard cubic feet, not considering deviation; E_v is the volumetric efficiency, as usually defined; P is the intake pressure, absolute; and D is the displacement.

In the term for volume, V_1, used for standard cubic feet at intake pressure, we evidently should include a term for deviation, for, if the intake gas is metered under standard conditions, or if it is measured under any other conditions and corrected for superexpansibility, as in modern metering practice, the actual displacement of the compressor at intake pressure P_1, *provided that deviation is present,* will be less than would be expected by the amount of the deviation from the ideal gas law, if the deviation factor is less than unity. If the deviation factor is above 1.00, the displacement will be correspondingly *greater.* Consequently

$$V_1 = C\,Y \tag{77}$$

where C is from equation 76 and Y is the deviation factor from equation 75. Consequently

$$V_1 = \frac{D\,E_v\,P\,Y}{14.73} \tag{78}$$

When there is no deviation, $V_1 = C$.

Compressor Horsepower. Since the actual displacement of the compressor will be less than expected, when a deviation factor less than 1.00 is used, the horsepower required for compression will be correspondingly less. For a deviation factor above 1.00, *more* horsepower is required.

Since $V_1 = C\,Y$, from equation 77, then equation 11 should be corrected to read:

$$HP = 0.1479\,Y\,C \log R \tag{79}$$

where C is in standard cubic feet per minute.

Equation 12 should read

$$HP = 0.10275\ Y\ C \log R \tag{80}$$

in which C is in thousands of standard cubic feet per 24 hours (M.c.f.).

Equation 16 should read

$$HP = 0.0643\ Y\ C \frac{n}{n-1} [R^{(n-1)/n} - 1] \tag{81}$$

in which C is in standard cubic feet per minute.

Equation 17 should read

$$HP = 0.0446\ Y\ C \frac{n}{n-1} [R^{(n-1)/n} - 1] \tag{82}$$

in which C is in thousands of cubic feet per 24 hours (M.c.f.).

Multistage Horsepower. When considering deviation, it is necessary to take each stage in multistage compression separately. Consequently, such horsepower equations as 34, 35, 41, 42, 48, and 49 cannot be used, when deviation is present, since the supercompressibility factors at the intake of the different stages of compression would necessarily be different from each other.

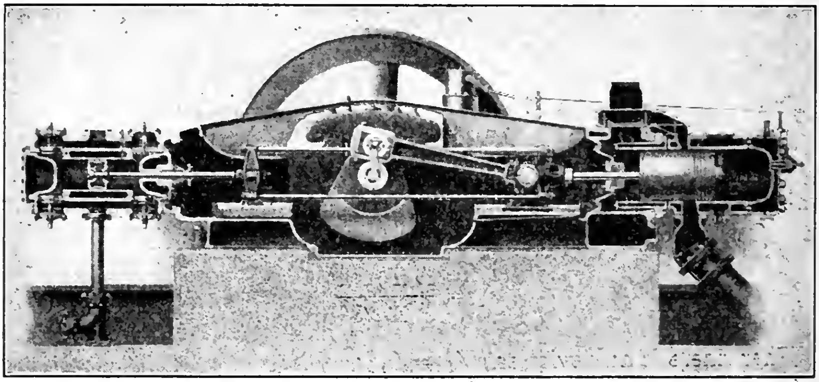


FIG. 21. Cooper-Bessemer type 12 single-gas-engine-driven compressor. (Made in 1935.)

Brake Horsepower. Equation 62, which gives brake horsepower per million cubic feet per 24 hours, should be corrected to read:

$$BHP \text{ per } M.M.c.f. = Y \left(\frac{44.6\ n}{n-1} [R^{(n-1)/n} - 1] + \text{Losses} \right) f(R) \tag{83}$$

All values of this equation in Table XIII should be multiplied by Y to correct for supercompressibility.

Maximum Ratio of Compression. Equation 24 contains only the clearance and n, and therefore requires no correction for deviation.

Indicated Horsepower. The indicator card and the mean effective pressure formulas give correct values of indicated horsepower. It is necessary, however, to correct the intake for deviation in computing capacities and horsepower per million.

Temperature Rise. Values of n taken from indicator cards, and used with Z from equation 69, will allow for deviation when computing temperature rise during compression.

Cylinder Sizes for Multistage Work. For equal work in each cylinder in multistage compression, the ratios of compression will not be the same in each stage if deviation from the ideal gas law is present.

As an example, let us take a two-stage machine with an overall ratio of compression of 16, working with $n = 1.20$. Assume a deviation factor Y (equation 75) as 0.90, for the high-pressure intake, and no deviation ($Y = 1.00$) for the low-pressure intake.

If the ratios of compression were equal in both stages, we should expect the high-pressure cylinder to do only 90 per cent as much work as the low-pressure, since the deviation is 0.90 for the high-pressure intake.

Taking R_0 as the ratio of compression for zero deviation and assuming that the horsepower curve from equation 62 is a straight line for the portion in which we are interested, then the general equation will be

$$\frac{R_0}{Y_1} \times \frac{R_0}{Y_2} \times \frac{R_0}{Y_3} \times \frac{R_0}{Y_4} \cdots = R_t \tag{84}$$

where the subscripts refer to stages 1, 2, 3, and 4; R_t is the overall ratio of compression; and $R_0/Y_1 = R_1$; $R_0/Y_2 = R_2$, etc.

Substituting in equation 84, $R_0 \times R_0/0.9 = 16$. Solving, $R_0 = R_1 = 3.80$, the ratio of compression of the first stage. $R_2 = R_0/0.9 = 4.22$, the ratio of compression of the second stage.

From equation 62 or Table XIII, the horsepower per million cubic feet would be 79.8 hp. for the first stage and $0.9 \times 87 = 78.3$ for the second stage.

From equation 32, assuming $R_1 = R_2 = \sqrt{16} = 4$, and a volumetric efficiency of 90 per cent in both stages,

$$R_1 = \frac{D_1}{D_2} = \frac{d_1^2}{d_2^2} \tag{32}$$

With a 12-in. low-pressure cylinder, making no allowance for deviation, equation 32 gives a high-pressure cylinder of 6-in. diameter.

In equation 73, for V we may substitute d^2, and we recall that $P_2/P_1 = R$. The terms for temperature may be cancelled if we assume perfect intercooling or the same intake temperature in both stages. The equation then becomes

$$\frac{d_1{}^2}{d_2{}^2} = \frac{Y_1 R_1}{Y_2} \tag{85}$$

Substituting in equation 85, $(12)^2/d_2{}^2 = 3.80/0.9$. Solving, $d_2 = 5.85$ in., the diameter of the high-pressure cylinder. In equation 85, using $d_2 = 5\frac{3}{4}$ in., we get $R_1 = 3.93$, and $R_2 = 4.075$.

With $R_1 = 3.93$, *BHP* per million for low-pressure cylinder is 82. With $R_2 = 4.075$, *BHP* per million for high-pressure cylinder is $84.8 \times 0.9 = 76.3$.

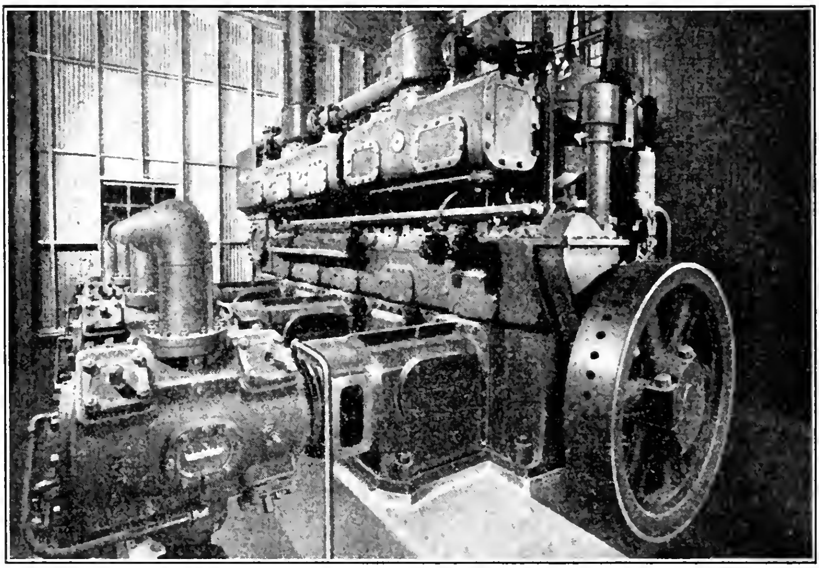


FIG. 22. Worthington type LTC gas-engine-driven compressor — 625 hp.

Using a 6-in. cylinder, from equation 85 we get $R_1 = 3.6$ and $R_2 = 4.45$.

With $R_1 = 3.6$, *BHP* per million for low-pressure cylinder is 76.3. With $R_2 = 4.45$, *BHP* per million for high-pressure cylinder is $90.5 \times 0.9 = 81.4$.

The problem is evidently a delicate one, if we insist upon exactly the same work in each cylinder. We could approximate the desired result with a cylinder $5\frac{7}{8}$ in. in diameter.

Problems for three- and four-stage compression may be solved by similar "cut-and-try" methods. The cylinder sizes should be determined in the ordinary way, without deviation, as outlined in Chapter IX, or by means of Charts 11, 14, and 16. Approximate ratios of compression can be computed from equation 84, and then a final check should be made on the brake horsepower.

Deviation, or Supercompressibility Data. The Bureau of Standards supercompressibility data gives δ as a function of PV/T. Other authorities base their values of deviation on PV/RT, which is substantially the same. As the values of Z and Y in equations 69 and 75 represent the quotient of two deviations factors, each multiplied by its corresponding absolute temperatures, it is evident that any slight discrepancies due to different standard pressures and temperatures for $\delta = 1.00$ will cancel out. Refer to equation 68.

Some textbooks,[1] however, give the deviation as a function of PV alone, and usually are based on $\delta = 1.00$ at one atmosphere and 0°C. In this case, the table of deviation factors will have a much greater range than when $\delta = PV/T$, the relationship being

$$\delta_{pv} = \delta_0 \frac{T}{T_s} \tag{86}$$

where δ_{pv} is the deviation factor based on $\delta = PV$; δ_0 is the deviation factor based on $\delta = PV/T$; T is the absolute temperature at which the deviation factor is taken; and T_s is the standard or base temperature. Usually, where δ is based on PV, the values of P are given in atmospheres.

Tables. Supercompressibility factors for air are given in Table XIV, and for methane in Table XV. Both these tables were replotted to Fahrenheit temperatures from Bureau of Standards data,[2] and are based on 14.7 lb. per sq. in. atmospheric pressure and 32°F.

Supercompressibility factors for natural gas, free of air and carbon dioxide, are given in Table XVI, based on orifice meter data from the California Natural Gasoline Association. In their original form, these data were designed for superexpansibility factors for orifice meters used in gas measurement. In the nomenclature of this text, the factors would represent $\sqrt{1 + \Delta}$. The orifice meter factors must be squared to obtain $(1 + \Delta)$, which is the form for correcting volumes measured by displacement meters. Values in Table XVI are reciprocals of the displacement meter factors. Table XVI represents considerable extrapolation in the higher ranges. This extrapolation was obtained by

[1] *Chemical Engineers' Handbook* by Perry, McGraw-Hill Book Co., 1934.

[2] *Miscl. Pub.* 71. 1925.

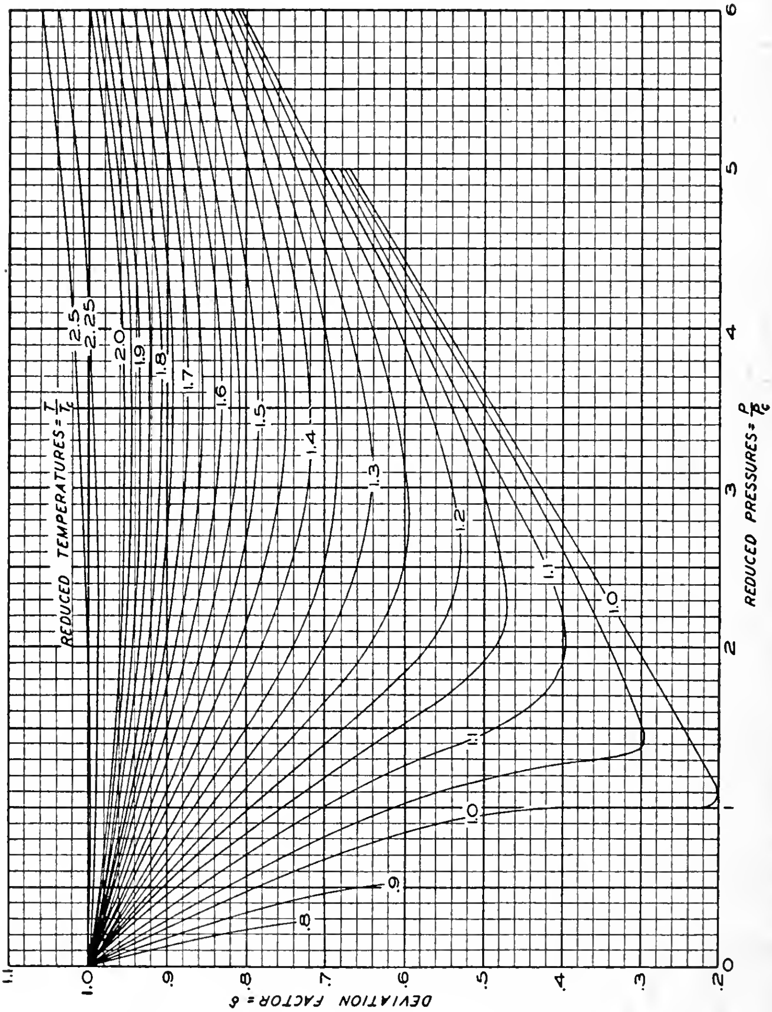

REDUCED TEMPERATURES = $\frac{T}{T_c}$
REDUCED PRESSURES = $\frac{P}{P_c}$
DEVIATION FACTOR = δ
2.5
2.25
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
.9
.8
0
1
2
3
4
5
6
1.1
1.0
.9
.8
.7
.6
.5
.4
.3
.2

Fig. 23.

plotting Δ on log paper with constant temperature and gravity, or with constant pressure and gravity.

High-Pressure Deviation. The deviation factors for natural gas in Table XVI are given in terms of specific gravity but do not extend beyond 800 lb. per sq. in. gauge. Supercompressibility data involving the specific gravity cannot satisfactorily be given for high pressures.

Above 500 lb. per sq. in. gauge, deviation data may be obtained by the pseudocritical point method, as outlined in a paper, "Deviation of Natural Gas at High Pressures," presented before the February, 1940, meeting of the California Natural Gasoline Association by J. N. Smith, of the Research and Development department of the Standard Oil Company of California.[1]

Deviation data are presented in terms of reduced temperatures (ratio of actual temperature to the critical temperature) and of reduced pressures (ratio of actual absolute pressure to critical pressure). Figure 23 gives the compressibility of the light paraffin hydrocarbons for the high-pressure range.

As natural gas consists of a mixture of a number of hydrocarbons, it is necessary to determine pseudocritical points to be used in Fig. 23 instead of the critical temperatures and pressures of pure hydrocarbons. These pseudocritical points are determined from an analysis of the gas by adding the critical temperature or pressure of each component in proportion to its mol fraction contained in the gas.

A typical example is given below:

Gas analysis, in mol per cent: carbon dioxide, 0.10; methane, 79.12; ethane, 9.92; propane, 5.86; isobutane, 0.93; N-butane, 2.25; pentane and heavier, 1.82. The Edwards balance specific gravity of the gas was 0.75. Required, the deviation for 2300 lb. per sq. in. abs. and 80° F.

Columns 1 and 2 give the mol per cent of the gas as shown by analysis. Column 3 is the molecular weight of the constituent gases. The residual, called "pentane and heavier," is taken as equivalent to N-hexane. Column 4 contains the products of mol per cent by molecular weight. The sum of the values in this column, 21.704, is the equivalent molecular weight of the gas. The specific gravity of the gas is the relation of this figure to the molecular weight of air, or 21.704/28.97 = 0.7491, which checks the Edwards balance reading of 0.75. Critical pressures and temperatures are given in Columns 5 and 7, and the products of their multiplication by the mol per cent are given in Columns 6 and 8. The total of Column 6, or 666.06 lb. prr sq. in. abs., is the pseudocritical pressure of the gas. The total of Column 8, or 406.72°F. abs., is the pseudocritical temperature.

[1] Also published in *California Oil World and Petroleum Industry*, first issue, February, 1940.

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
Gas Analysis	Mol Per Cent	Molecular Weight		Critical Pressure (abs.)		Critical Temp. (°F. abs.)	
Carbon dioxide....	0.10	44.00	0.044				
Methane.........	79.12	16.03	12.683	673	533.15	344	272.52
Ethane...........	9.92	30.05	2.981	717	71.12	550	54.56
Propane..........	5.86	44.06	2.582	632	37.03	664	38.91
Isobutane........	0.93	58.08	0.540	544	5.06	733	6.82
N-Butane........	2.25	58.08	1.307	529	11.90	767	17.26
Pentane (+) ..... (taken as N-hexane)	1.82	86.11	1.567	434	7.80	915	16.65
	100.00		21.704		666.06 lb. abs.		406.72 °F abs.

NOTE. The law of corresponding states, upon which the pseudocritical method is founded, applies only to hydrocarbons. It is customary, therefore, to combine the carbon dioxide in the gas with the methane to determine the pseudocritical points. Usually the possibility of error by this method is very small.

The deviation is required for 2300 lb. per sq. in. abs. and 80°F. The reduced pressure is $2300/666 = 3.45$; the reduced temperature is $(80 + 460)/407 = 1.325$. Using Fig. 23, the deviation is 0.670.

As a further check, the deviation is required for the above gas at 400 lb. gauge and 200°F. Reduced pressure is $414.7/666 = 0.622$; reduced temperature is $660/407 = 1.62$. From Fig. 23, deviation is 0.955. Compare with Table XVI.

REFERENCES ON SUPERCOMPRESSIBILITY

COPE, LEWIS, AND WEBER, *Ind. Eng. Chem.*, vol. 23, p. 887, August, 1931.

LEWIS AND LUKE, *Ind. Eng. Chem.*, vol. 25, p. 725, 1933.

W. B. KAY, *Ind. Eng. Chem.*, vol. 28, p. 1014, 1936.

KVALNES AND GADDY, *J. Am. Chem. Soc.*, vol. 35, p. 394.

PROF. GEORGE GRANGER BROWN, "Compressibility of Pure Gases," *Petroleum Engineer*, January, 1940.

PROF. GEORGE GRANGER BROWN AND DYSART E. HOLCOMB, "Compressibility of Gaseous Mixtures," *Petroleum Engineer*, February, 1940.

BROWN, SAUNDERS, AND SMITH, *Ind. Eng. Chem.*, vol. 24, p. 513, 1932.

SAGE, LACEY, AND SCHAAFSMA, *Ind. Eng. Chem.*, vol. 26, p. 214, 1936.

SAGE AND LACEY, paper presented at Twentieth Annual Meeting, American Petroleum Institute, Nov. 17, 1939.

CHAPTER XIV

COMPRESSORS

Like any other piece of industrial machinery, the compressor has undergone a number of evolutionary steps in adapting itself to the service required of it. Although the general design of the compressor cylinder is still the same and will probably remain so until superseded by some other fundamentally different arrangement, there have been many changes in regard to valves, pistons, crankshafts, and prime movers.

The early compressors were used largely for air compression or in refrigeration, and were usually equipped with mechanically operated valves. An early Laidlaw-Dunn-Gordon compressor,[1] having Corliss-type valves, is shown in Fig. 24.

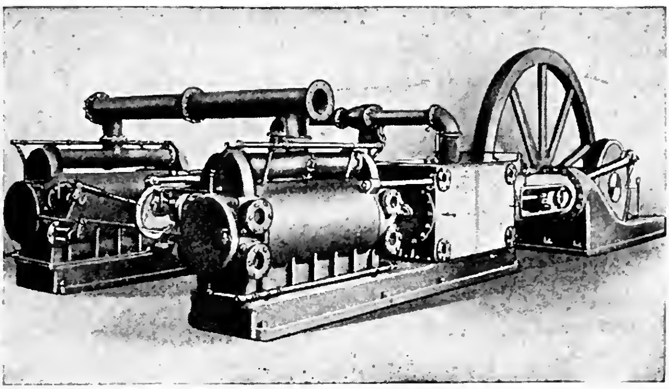


FIG. 24. Early Laidlaw-Dunn-Gordon air compressor, with outside mechanically operated valves. (Built about 1900.)

Mechanically operated valves have the inherent disadvantage that they must be set to open or close at exactly the right point, or there will be loss of power, shown by a small bulge or hump in the indicator card. This condition would resemble the sample indicator curve shown

[1] From the *One Hundred Year Anniversary Volume* of the Worthington Pump and Machinery Co., 1940.

in Fig. 13, with stiff valve springs. The operating conditions are usually very constant in refrigeration plants and in most air compressors, a fact which favored the use of mechanically operated valve for a number of years.

Where there were varying loads, the disadvantages of the mechanically operated valve soon became apparent, and various types of valves were used in an effort to overcome the difficulty. On modern compressors, the valves are actuated by pressure alone and are entirely automatic. Figure 25 shows the Worthington feather valve, the operating principle of which may be taken as typical of modern practice. Essentially, this valve consists of long narrow ports, covered by thin metal strips. In the discharge valve, for instance, the line or reservoir pressure into which the compressor is pumping forces these strips to seat over the ports, thus closing the valve during the compression stroke. When the pressure in the cylinder reaches that of the discharge line, the little strips are pushed away from their seats and the valve opens.

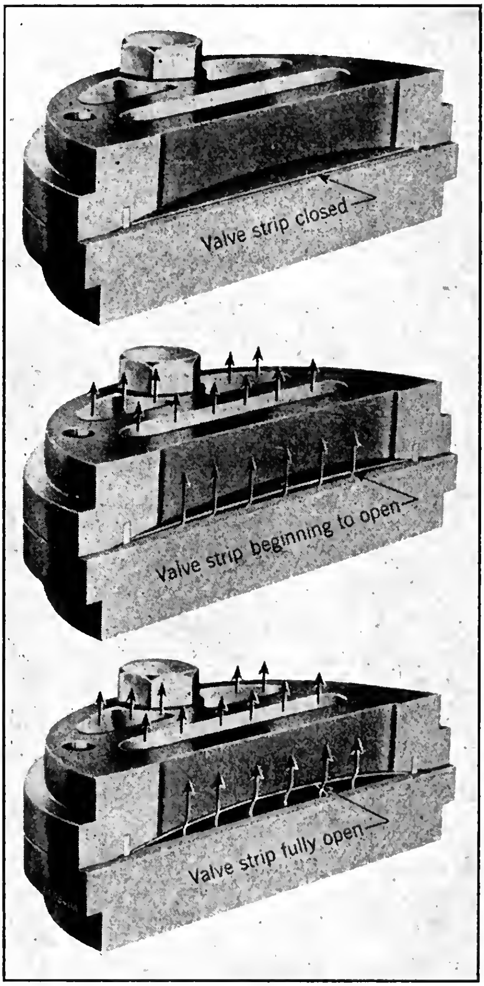


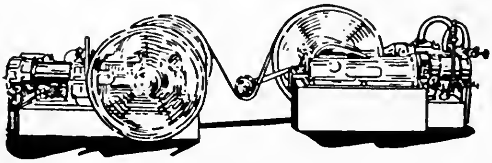
Fig. 25. Worthington feather valve.

Other manufacturers use the strip in modified form for low-pressure work, but for medium and high pressures, use circular disc valves, seated by small springs.

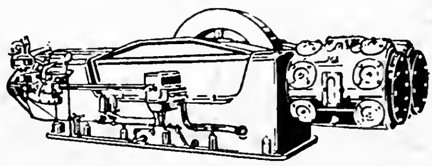
In its evolution, the compressor has inclined towards higher speeds and compactness, requiring smaller floor space and smaller foundations.

In early installations, the prime mover and the compressor were built separate and connected by a belt drive, sometimes resulting in units as much as 30 or 40 ft. long. See Fig. 26 (*A*).[1]

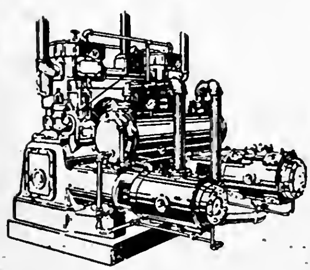
The next step was direct-connected units (*B*) in which there would be a horizontal power cylinder at one end and a compressor cylinder at the other. The usual form was a " four-corner " type, with two power cylinders and two compressor cylinders, and a large flywheel in the center. Units of 1500 hp. and above were common in this type and are still being widely used.



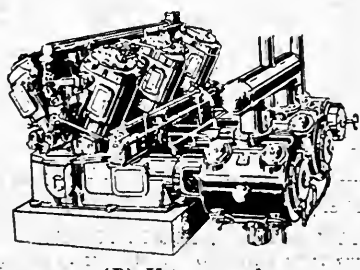
(*A*) Belt drive.



(*B*) Horizontal cylinders, direct connected.



(*C*) Vertical power cylinders, 90° angle type.



(*D*) V-type angle.

FIG. 26.—Evolution of the compressor.

Tendency of late has been towards somewhat higher speeds and compactness, which resulted in the angle type of machine (*C*). Here we have a 90° installation, with vertical power cylinders and horizontal compressor cylinders.

A variation of this design is the gas engine drive compressor (*D*), in which the compressor cylinder is horizontal and the power cylinders

[1] Furnished by the Ingersoll-Rand Co., 11 Broadway, New York.

form a vertical " V." There are usually more power cylinders than compressor cylinders for this type of machine.

Other types of compressors depend largely upon the kind of prime mover and the size of the installation. Synchronous motors are often mounted as the flywheel on horizontal units. There are various forms of electric drives for the smaller units, using the V-type belt connection. Portable units for air compression are often seen with vertical gas engine and compressor cylinders arranged in line.

Table XVII gives a variety of data on a representative list of compressor cylinders.[1] Of late, manufacturers have shown a tendency to increase valve areas, so that some of the values in the table may be low for new installations.

Clark Bros. Co., Inc., of Olean, New York, manufactures an extensive line of oil-field equipment. Emphasis is placed on the manufacture of natural-gas-engine direct-driven compressors in sizes from 40 to 1000 hp., but other products include compressors with Diesel engine, steam engine, and electric motor drive; expander units, multicylinder gas and Diesel engines for drilling or generator service; and oil-well pumping engines. Though some four-cycle engines are included among its products, the company is best known for the excellence of its two-cycle gas-engine designs. A pioneer in the use of the fuel injection system for two-cycle gas engines, it has developed this type to a point where its thermal efficiency is said to equal that of the best four-cycle designs. At the present time, its most successful product is a two-cycle natural-gas-engine direct-driven compressor unit, which is made in twelve sizes, ranging from 165 to 1000 hp.

Cooper-Bessemer Corp., of Mount Vernon, Ohio (established in 1833), in addition to its line of horizontal and vertical gas and Diesel engines for direct power take-off, manufactures many types of single and twin cylinder, single and twin tandem, horizontal compressors. Within the last few years, it has introduced its Type G-MR " angle " and Type G-MV " V-angle " units. Both are gas-engine driven, the Type G-MR employing vertical power cylinders and horizontal compressor cylinders, while Type G-MV has its power cylinders in a vertical " V " arrangement, with compressor cylinders horizontal. In common with the general trend of the industry, these machines are designed for compactness and ease in installation. The Type G-MV units, which are rated at 100 hp. per cylinder, are built in four-, six-, eight-, and ten-cylinder sizes and are equipped with oil-cooled pistons, streamlined scavenging, separate scavenging pistons for each engine crankthrow,

[1] Based on data compiled by Lyman F. Scheel.

Silent-Scot fuel injection, full-pressure lubrication, and precision-type bearings throughout.

Ingersoll-Rand, 11 Broadway, New York, has a complete line of compressors for air, gas, and ammonia, consisting of more than 1000 sizes and types, ranging from $\frac{1}{4}$ to 3000 hp. Its smallest machines are the type T-30 air compressors, which are self-contained, air-cooled, single-acting units with V-belt drive from electric motors or gasoline engines. This type is rated from $\frac{1}{4}$ to 15 hp. and is largely used in filling stations and garages. The type PRE, from one to six stages, is built from 200 to 3000 hp. and for pressures of 5 to 5000 lb. per sq. in. These large machines are driven by synchronous motors on the flywheel and are equipped with five-step clearance control for load regulation. In recent years, the company has introduced the XVG angle type compressor for oil-field service, in sizes from 65 to 650 brake horsepower. They are direct connected to four-cycle V-type gas engines, and are designed for use in gas lift, repressuring, recycling, and for general gas gathering, gas transmission, and refinery work. They require only about half the floor space of horizontal machines of the same capacity. The Ingersoll-Rand line also includes centrifugal blowers, from one- to six-stage, the largest sizes requiring 12,000 hp.

The Worthington Pump and Machinery Corp., of Harrison, N.J., has recently issued its *Hundred Year Anniversary volume* (1840–1940) giving an interesting history of the evolution of many forms of pumps, compressors, and other machinery during the last century. In 1900, a great extension of the air and gas compressor business was under way, and in 1902, the company began to manufacture horizontal gas-engine-driven gas compressors. Worthington perfected the Cincinnati valve gear, which was designed to operate the inlet and discharge valves mechanically. In 1915, the company developed the " feather valve," modifications of which are still used in the majority of compressors now built. See Fig. 25. The Worthington Corp. now manufactures a complete line of compressors and vacuum pumps for a variety of purposes, with gas engines, steam, and motor drives, ranging in size from small portable units, to such installations as the two 2500-hp. motor-driven compressors at the Calumet station of the People's Gas Light and Coke Co. in Chicago. In recent years, the company has successfully introduced a line of two-cycle angle-gas-engine-driven compressors for use in the petroleum industry and for natural-gas transmission work.

CHAPTER XV

COMPRESSOR PLANT

The Memphis Natural Gas Co.'s compressor station[1] near Wilmot, Ark., is a good example of modern compressor plant design. The plant is located near the Arkansas-Louisiana line, four miles east of Wilmot, Ark., on the 18-in. gas trunk line supplying gas for domestic uses in Memphis, Tenn., from the oil and gas wells near Monroe, La. For interior view see Fig. 27.

The main compressor plant is composed of five Clark super-two-cycle right-angle compressors of 600 hp. each, at a rated speed of 300 r.p.m. Each unit consists of a six-cylinder, two-cycle, 14-in. by 14-in. right-angle vertical gas engine, direct connected to four 8-in. horizontal compressor cylinders, with intake from 150 to 250 lb. per sq. in. gauge, and discharge pressure from 400 to 450 lb. gauge. Load variation is controlled by clearance pockets on each cylinder.

The capacity of each unit can be varied from 11 to 23 million standard cubic feet per 24 hours. Each machine has two horizontal double-acting scavenging cylinders of 20-in. diameter, which provide scavenging air at about $3\frac{1}{2}$ lb. pressure and discharge into a common header.

All main and auxiliary engines have overhead exhausts, which are insulated inside the building with Fluor-type jackets. In this design of jacket, an air space between the outside of the exhaust and the jacket forms a vent that carries off the heated air through the roof.

The gauge board in the main compressor building has two indicating gauges for gas suction and discharge pressures; two recording thermometers for suction and discharge temperatures; and three indicating gauges for gas fuel pressure, jacket water pressure, and starting air pressure.

The water supply of the plant comes from a well 630 ft. deep. Water is raised to the surface by gas lift, and stored in a 30,000-gallon elevated steel water tank.

Of the five water pumps in the water-cooling system, two take suction from the hot well and pump over the top of the cooling tower; two more

[1] See "Memphis Natural Gas Co.'s Wilmot Compressor Station" by Wallace and Higman, *Petroleum Engineer*, February, 1938.

take suction from the base of the cooling tower and pump through the gas coolers and engine jackets; the remaining pump is a spare, connected to replace either of the others as required.

The elevated water tank floats on the discharge of the water pumps that pump through the gas coolers and engine jackets. Raw water from the well discharges into a 6-in. line that goes to the hot well. From here the water is pumped over the top of the cooling tower, then drawn from the cooling-tower basin and pumped through the gas coolers and engine jackets. It then returns to the hot well, from which it is recirculated over the cooling tower.

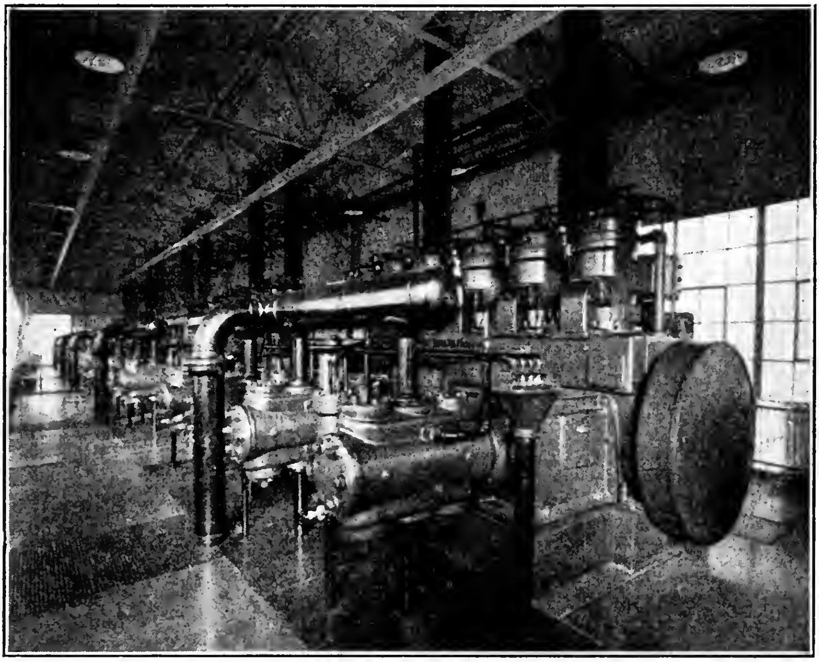


FIG. 27. Clark 600-hp. two-cycle model RA-6 gas-engine-driven compressors at Memphis Natural Gas Co. Plant, Wilmot, Arkansas.

The cooling tower, a Fluor Aerator type, has a cooling capacity of 975 gallons per minute from 98° to 70°F., based on 45° wet bulb and a 3-mile-per-hour wind velocity.

The gas coolers are of the enclosed type, three coolers being provided on the gas discharge headers from the compressors, each designed for a maximum working pressure of 450 lb. per sq. in. in the shell and 100 lb. in the tubes. Their capacity is sufficient to cool 88 million standard

cubic feet of compressed gas to 90°F. under winter operating conditions, when the gas is compressed from 225 to 450 lb. gauge.

Gas is taken into the plant from the 18-in. trunk line through an 18-in. suction header. Each machine is supplied through an 8-in. suction line which connects with a 12-in. manifold at the compressor. Gas is delivered to the cylinders from the manifold by 4-in. lines. From the

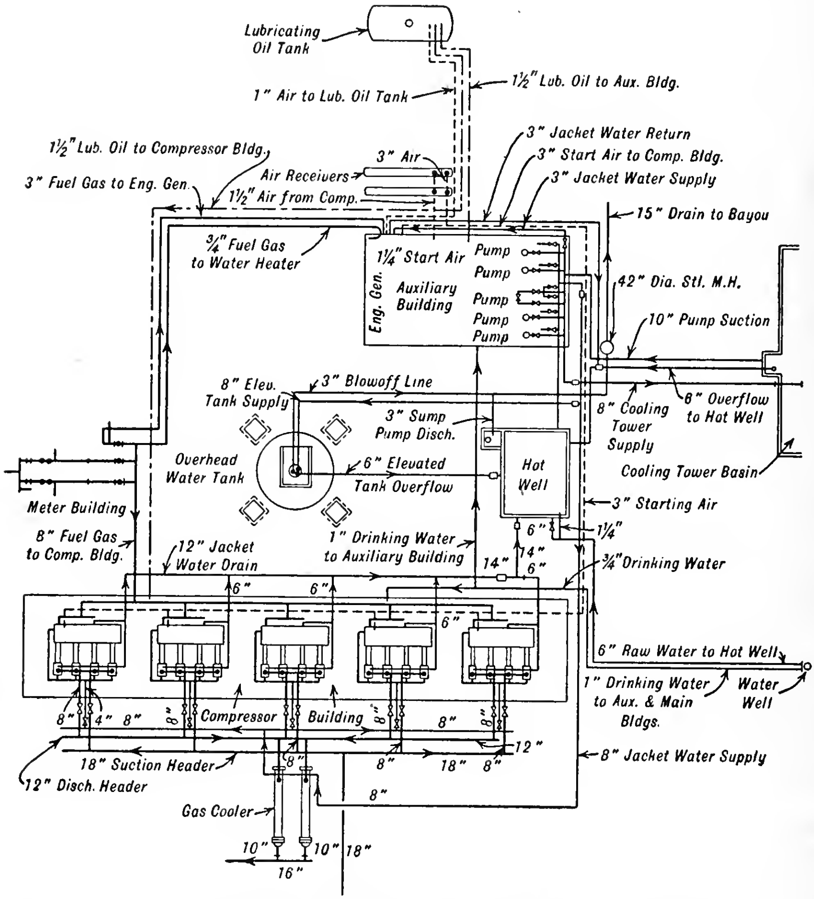


Fig. 28. Flow sheet of Memphis Natural Gas Co. Compressor Station, Wilmot, Arkansas.

compressor cylinders the gas is discharged through 4-in. outlets into a 12-in. manifold below the floor, thence through an 8-in. discharge line to another 12-in. manifold outside the building, and from there through three 10-in. risers to the three gas coolers. From the coolers the gas is

discharged through 10-in. lines into the 16-in. station discharge line, which feeds into the trunk line.

In the auxiliaries building, there is an electric generator unit which consists of a 75-kw. 113-ampere 480-volt generator, driven by a vertical multicylinder four-cycle gas engine of sufficient size to carry the full lighting and power load of the plant.

The centrifugal water pumps, driven by electric motors, have a capacity of 490 gallons per minute at a rated speed of 1750 r.p.m.

The starting-air compressors are of the V-type. One is direct driven by an induction motor with automatic pressure-actuated start-stop switch, and the other is driven by a gas engine through a V-belt drive. Each compressor has a capacity of 30 cu. ft. per min., with atmospheric intake and 250 lb. discharge. Compressed starting air is stored in two receivers, 24 in. in diameter and 20 ft. long. Air from the receivers is delivered to the main building through a 3-in. line, with $1\frac{1}{4}$-in. lines branching off to each engine. The engines have an air-starting valve on each cylinder, a valve in the cylinder head, and pilot valves on each control panel.

Fuel gas is taken from the suction header through a 2-in. line to the meter house. Here the line branches into two meter runs, through each of which the gas is measured and the pressure reduced, first to 90 lb. and then from 90 lb. to 27 lb. per sq. in. The gas then goes to a header in the compressor building where the pressure is then reduced at each engine to 17 lb. gauge.

From the meter house an 8-in. fuel gas line leads to the main compressor building, and a 3-in. line to the auxiliaries building.

A flow sheet of the compressor plant is given in Fig. 28.

CHAPTER XVI

GAS LIFT AND PRESSURE MAINTENANCE

For many years, the petroleum and natural-gas industries have been important users of compressors, largely for transmission purposes and for pulling a vacuum on the low-pressure wells in order to increase the yield of oil and gas.

Subsequent to about 1926, however, petroleum engineers, beginning to fear a shortage of oil, considered methods of checking the natural decline of the pressure in the oil sands as the oil and gas were removed. Practically all the methods suggested called for the introduction of high-pressure gas into the producing formations, either to raise the oil to the surface or to increase the yield of adjoining wells.

Figure 29 shows a typical flow sheet of an active oil field in which a small amount of drilling of new wells is still going on. The oil and gas from a low-pressure well are taken through a gas trap, or oil and gas separator, from which the oil is piped off to the tanks. The wet gas from the trap is pulled in by the low-pressure compressors, which may or may not operate at a vacuum, depending on the pressures at the wells. After being cooled in a cooling tower, the gas from the low-pressure compressors is usually taken to absorbers, where the natural or casinghead gasoline is removed. The resulting dry gas is then removed by the high-pressure compressors and compressed to about 300 to 500 lb. per sq. in., being distributed to the individual wells through a gas lift manifold or header. The gas lift line to the low-pressure well is usually connected to the casing of the oil well, the gas being forced downward in the annular space between casing and tubing. A spray of oil and gas is brought to the surface of the ground through the tubing and is piped to the gas trap, where the oil is separated from the gas and the cycle completed.

In high-pressure wells, the wet gas from the trap is by-passed around the low-pressure compressors and goes directly to the absorption plant. From this point on, the gas follows the same course as the gas from a low-pressure well.

When high-pressure gas is forced into a well, but no oil or gas is taken from it, the process is known as gas injection, gas storage, or repressuring.

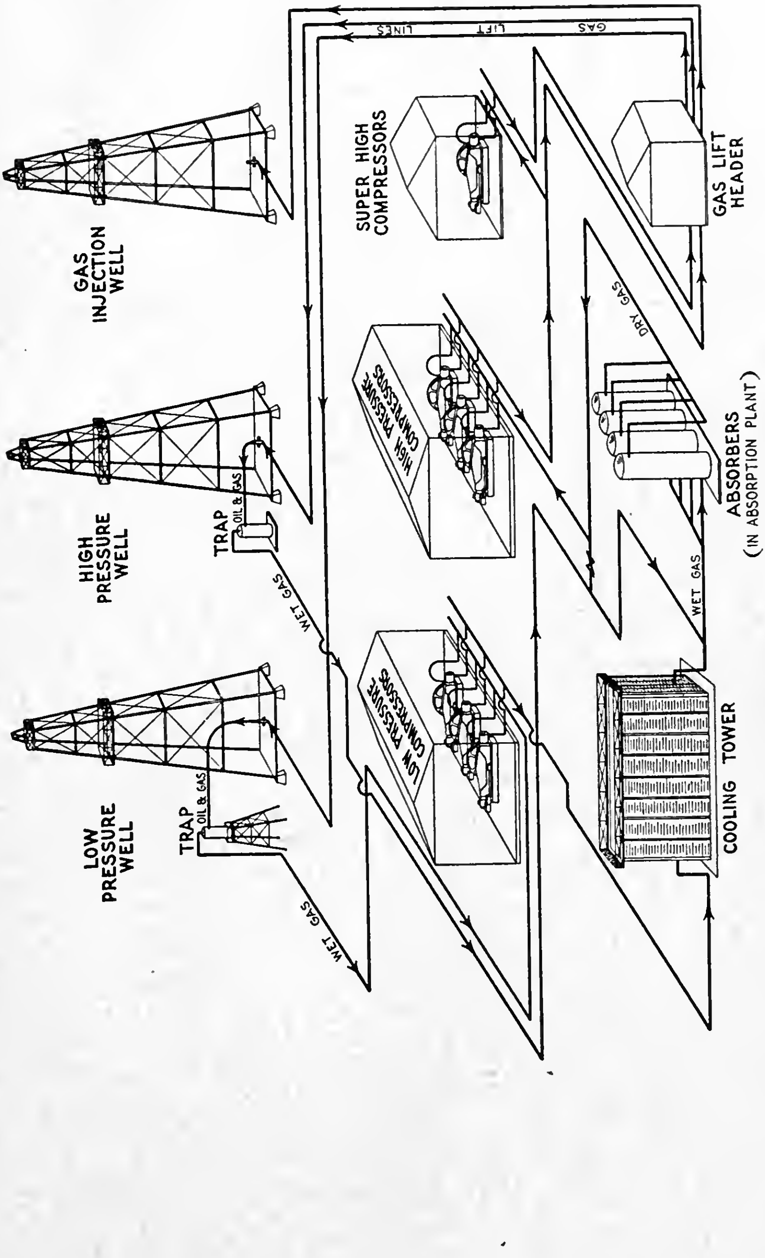

GAS INJECTION WELL
HIGH PRESSURE WELL
LOW PRESSURE WELL
TRAP
OIL & GAS
TRAP
OIL & GAS
WET GAS
WET GAS
GAS LIFT LINES
SUPER HIGH COMPRESSORS
HIGH PRESSURE COMPRESSORS
LOW PRESSURE COMPRESSORS
GAS LIFT HEADER
DRY GAS
ABSORBERS
(IN ABSORPTION PLANT)
WET GAS
COOLING TOWER

Fig. 29.

Gas lift is essentially a dynamic process, the oil being brought to the surface not by a static lift, as the name might indicate, but by the velocity of the gas in the tubing. When gas lift is being started, or when the well has been idle for some time, the fluid level frequently rises several hundred feet above the bottom of the tubing. When this occurs, it is necessary to provide super high, or booster, compressors to supply

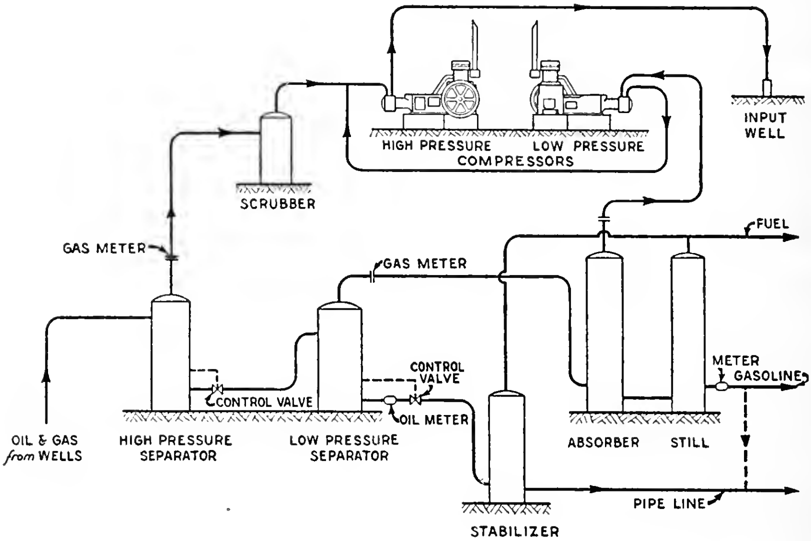


Fig. 30.

enough pressure to overcome the static head which results when the oil falls in the casing and rises in the tubing. As the column of liquid is forced out through the tubing, the pressure falls and gas iift begins.

Variations of the original gas lift process are variously known as repressuring, recycling, or pressure maintenance. The exact methods employed, and the pressures to be maintained, are largely problems of petroleum engineering, and concern us only as regards the compressors to be used.

Since about 1935, there has been a spectacular increase of the recycling and repressuring process in the Mid Continent and Gulf Coast oil fields, with the resulting installation of an enormous horsepower in high-pressure compressors. A typical flow diagram of a high-pressure field is shown in Fig. 30.[1] In the so-called gas-distillate fields, where little

[1] From a paper by E. O. Bennett, read before the American Petroleum Institute, Wichita, Kansas, May 24, 1938, entitled, "Pressure Maintenance."

or no fluid is produced at the casinghead, the distillate or condensate is recovered by partial expansion, often accompanied by cooling, either by atmospheric cooling towers or by some form of refrigeration. Before being returned to the formation, the gas may or may not be treated in an absorption plant.

The high pressures employed in recycling and repressuring, which sometimes reach 4500 lb. per sq. in. have presented many new problems to the compressor manufacturer, as well as to the manufacturer of valves, fittings, and other equipment.

The compressors for high-pressure work must be fitted with special high-pressure cylinders, which are necessarily of small diameter. The ratios of compression rarely exceed 3. For unusually high pressures, the cylinders are sometimes made single-acting, of the ram type. A typical installation may be cited as the Clark 600-hp. angle-type units at the recycling plant of the Corpus Christi Corp., near Corpus Christi, Texas. Each unit has six 14-in. by 14-in. power cylinders, with four $3\frac{1}{2}$-in. bore and 14-in. stroke forged-steel compressor cylinders. Typical operating pressures are from 1200 to 3100 lb. per sq. in. gauge.[1]

BIBLIOGRAPHY ON REPRESSURING AND RECYCLING

" Gas Recycling and Distillate Recovery " a series of articles reprinted by Clark Bros. Co., Inc., Olean, N. Y. The article by F. W. Laverty, director of technical service of the Clark Bros. Co., contains an extensive bibliography.

E. O. Bennett and C. R. Williams, " Crude Stabilization and Return of Residue Gases to Wells " paper presented before Natural Gasoline Association of America at Tulsa, Okla., May, 1937.

Mitchell Tucker, " Cook Field Exemplified Value of Pressure Maintenance," *Oil and Gas J.*, Jan. 27, 1938.

H. N. Wade, " Pressure Maintenance for Economic Distillate Structure Production is Essential," *Oil Weekly*, March 27, 1939.

Brad Mills, " Distillate Fields Bring New Production Phase," a series of articles in *Oil Weekly*, July 29, 1940, p. 53; Aug. 5, 1940, p. 13; Aug. 12, 1940, p. 19; Aug. 19, 1940, p. 28.

[1] For complete description of a recycling plant, see " Recycling Plant Improves Efficiency of Production in Stratton Field " by E. E. de Back *Petroleum Engineer*, March, 1939.

CHAPTER XVII

FLOW OF GAS IN PIPE LINES

A number of formulas for pressure drop of gas flowing in pipe lines have been published,[1] most of them being largely empirical in nature. Recent works on the flow of fluids in pipes usually introduce Reynolds' criterion,[2] which applies to both gases and liquids and is practically indispensable for dealing with liquids other than water.

As the Reynolds number includes an expression for viscosity which is difficult to apply in mixtures of gases such as natural gas, practical considerations recommend the use of Weymouth's equation, first proposed by Thomas R. Weymouth in 1912.[3] Recent investigations with this formula, using Reynolds number to form a correction factor,[4] show that results seem to lie on the conservative side, with enough leeway to allow for unforeseen losses, especially those caused by obstructions in the line, condensation, or pulsation. The Weymouth formula is not recommended[5] for pressures above 3000 lb. per sq. in.

Since the Weymouth equation is largely empirical in nature, its complete derivation cannot be given. It is based, however, on the Chezy formula,[6] given in all textbooks of hydraulics:

$$Q = A\,C\sqrt{2g\,R\,S} \tag{87}$$

where Q is the volume passing through the pipe; A is the cross-sectional area of the pipe; C is a dimensional constant; g is the acceleration of gravity, which is usually taken as 32.2 ft. per sec. per sec.; R is the

[1] See *Natural Gas* by Lester C. Lichty, p. 371, John Wiley & Sons, 1924.

[2] Now given in most works on thermodynamics. We quote only two: *Principles of Chemical Engineering* by Walker, Lewis, and McAdams, p. 73, McGraw-Hill Book Co., 1923; *Thermodynamics* by Lichty, p. 207, McGraw-Hill Book Co., 1936.

[3] *Trans. Am. Soc. Mech. Engrs.*, vol. 34, 1912.

[4] By Benjamin Miller of the Cities Service Co., in *Gas*, November, 1937. See also "Factors Influencing Flow of Natural Gas Through High Pressure Transmission Lines," by Berwald and Johnson, U. S. Dept. of Commerce, Bureau of Mines, *Report of Investigation No. 3153*, Washington, 1931.

[5] See "Crude Stabilization and Return of Residue Gases to Wells" by Bennett and Williams, a paper presented before the annual meeting of the Natural Gasoline Association of America at Tulsa, Okla., May, 1937.

[6] *Hydraulics* by Joseph N. LeConte, p. 88, McGraw-Hill Book Co.

hydraulic radius of the pipe, or the ratio of the cross-sectional area to the wetted perimeter, which equals $d/4$ for circular pipes; and S is the hydraulic slope, or head lost divided by the length of pipe, which is H/L.

For the head lost, H, we may substitute its equivalent, P/γ, where γ is the weight of the flowing fluid in pounds per cubic foot. It is convenient, however, to express γ in terms of G, the specific gravity of the gas, based on air = 1.00.

Substituting, we now have

$$Q = C\,\frac{\pi d^2}{4}\sqrt{\frac{2g}{L}\,\frac{P}{G}\,\frac{d}{4}}$$

Combining all the constants together with C, and bringing d^2 inside the radical,

$$Q = C\sqrt{\frac{P}{L}\,\frac{d^5}{G}}$$

It is now necessary to find a more exact expression for P, which represents the varying pressure within the pipe. We assume that all the drop in pressure is due to pipe friction, neglecting any changes in temperature. As gas is a compressible fluid, we take each minute change of pressure as being proportional to the instantaneous pressure at the point where the pressure drop occurred. Integrating $P\,dP$ between any two points along the pipe line for the pressure drop,[1] we have

$$P = \int_1^2 P\,dP = \frac{P_1{}^2 - P_2{}^2}{2}$$

We now have the equation in the general form,

$$Q = C\sqrt{\frac{(P_1{}^2 - P_2{}^2)\,d^5}{L\,G}}$$

Weymouth's experiments showed that pipe friction also varies as the one-third power of the pipe diameter.

Inserting $d^{1/3}$ and necessary dimensional constants, the equation becomes

$$Q = 48.76\sqrt{\frac{(P_1{}^2 - P_2{}^2)\,d^{5.33}}{L\,G}} \qquad (88)$$

where Q is the line capacity in thousands of standard cubic feet per 24

[1] The same result is obtained by assuming the head for a compressible fluid to vary as the product of the pressure drop by the average pressure. Or $H \doteq (P_1 - P_2) \times \frac{1}{2}(P_1 + P_2) = \frac{1}{2}(P_1{}^2 - P_2{}^2)$, as above.

hours (M.c.f.); P_1 and P_2 are the initial and final pressures of the gas in pounds per square inch absolute; d is the internal diameter of the pipe in inches; L is the length of the line in feet; and G is the specific gravity of the gas based on air = 1.00. Equation 88, known as the Weymouth formula for pipe-line pressure drop, is based on a flowing temperature of 60° F. If it is desired to use any other temperature, equation 88 may be written

$$Q = 1112\sqrt{\frac{(P_1^2 - P_2^2)\,d^{5.33}}{L\,T\,G}} \tag{89}$$

where T is the absolute temperature of the gas in the pipe.

Numerical computations involving equations 88 and 89 are somewhat tedious, principally because of the pressure-drop radical, $\sqrt{P_1^2 - P_2^2}$, involving the initial and final absolute pressures. To simplify matters, let U equal the per cent pressure drop (expressed as a decimal), based on the initial absolute pressure in the pipe. Then

$$U = \frac{P_1 - P_2}{P_1}$$

$$UP_1 = P_1 - P_2$$

$$P_2 = P_1(1 - U)$$

$$\sqrt{P_1^2 - P_2^2} = \sqrt{P_1^2 - P_1^2(1 - U)^2} = P_1\sqrt{1 - (1 - U)^2}$$
$$= P_1\sqrt{U(2 - U)}$$

Substituting in equation 88,

$$Q = 48.76\,P_1\sqrt{\frac{U(2 - U)\,d^{5.33}}{L\,G}} \tag{90}$$

Data for the solution of equation 90 are given in Table XVIII. For ease in calculation, the equation has been split up as follows:

$$Q = \left[\sqrt{\frac{1}{L}}\right]\left[48.76\sqrt{\frac{1}{G}}\right] d^{8/3}\ \left[\sqrt{U\,(2 - U)}\right] P_1 \tag{91}$$

As P_1 represents absolute pressure, it must be remembered that it is necessary to add 14.73 lb. per sq. in. to the gauge pressure in making calculations.

When the initial pressure is below about 30 lb. gauge, some difficulty will be encountered from the fact that vacuums may be reached in the value of P_2. Table XIX gives final pressures (P_2) for various percentages of pressure drop with initial pressures of 30 lb. gauge or less.

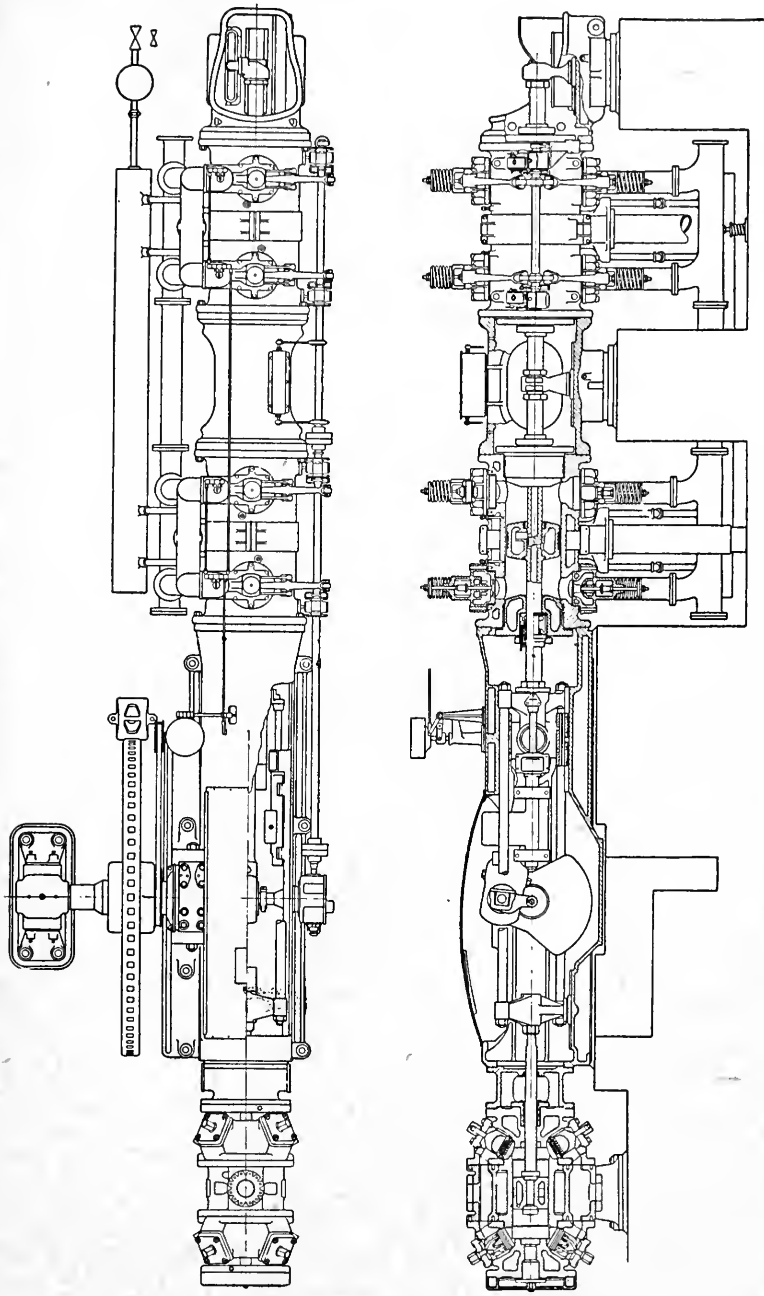


FIG. 31. Worthington double-acting tandem gas-engine-driven compressor.

Tables XX to XXIX, inclusive, have been provided for the solution of equations 88 to 91 for pipe sizes from 2- to 20-in. and for lengths from 100 to 100,000 ft. A gravity of 0.75 is assumed for the gas. To secure numerical results, for pipe capacities in thousands of cubic feet per 24 hours, it is only necessary to multiply the tabular values by the absolute initial pressure of the gas.

Complex Pipe Lines. Equations 88 to 91 give the pressure drop in a pipe of uniform diameter. To observe the effect of length and pipe diameter, let us take two pipes of different sizes and lengths, but with the same quantity of gas flowing, and with the same pressure drop and specific gravity. Writing equation 88 for both pipes, and making the two expressions equal to each other, since Q is the same in both, we find that everything cancels out except

$$\sqrt{\frac{d_1^{5.33}}{L_1}} = \sqrt{\frac{d_2^{5.33}}{L_2}} \tag{92}$$

Squaring both sides of the equation and solving for the length of the first pipe,

$$L_1 = L_2\left[\frac{d_1}{d_2}\right]^{5.33} \tag{93}$$

where L is in feet and d is in inches.

Thus, if we have a pipe line consisting of 4000 ft. of 6-in. pipe and 500 ft. of 4-in. pipe, we can reduce the 500 ft. of 4-in. pipe to an equivalent length of 6-in. and treat the whole if it were all 6-in.

Substituting in equation 93,

$$L_6 = L_4\left[\frac{d_6}{d_4}\right]^{5.33} = 500\left[\frac{6.065}{4.026}\right]^{5.33} = 500 \times 8.89 = 4450 \text{ ft.}$$

The entire line would therefore be equivalent to $4000 + 4450 =$ 8450 ft. of 6-in. pipe. Any number of sections of pipe of different sizes, connected in series, can thus be reduced to equivalent lengths, and the total equivalent length of pipe can be treated as if it were of uniform size.

Loop Lines. When two pipes are connected at the ends to form a loop, we must first reduce them to the same equivalent length, if both sides of the loop are not equal. This can be done by rewriting equation 93 and solving for the diameter,

$$d_1 = d_2\left(\frac{L_1}{L_2}\right)^{3/16} \tag{94}$$

Let us consider 4000 ft. of 3-in. pipe and 2500 ft. of 4-in. pipe forming a loop and connected at the ends. We can allow for a length of 2500

ft. or 4000 ft. or any other length desired, but it is usually easier to reduce one side of the loop to the same length as the other, and then no calculations are needed for the remaining side. Let us use the shorter side, 2500 ft. The problem is to find the pipe size, 2500 ft. of which will have the same pressure drop as 4000 ft. of 3-in. Substitute in equation 94,

$$d = 3.068 \left[\frac{2500}{4000}\right]^{3/16} = 3.068 \times 0.9166 = 2.81 \text{ in.}$$

We now have a loop 2500 ft. long on both sides, consisting of 4- and 2.81-in. inside diameter pipe.

If we should write equations for the capacities of both sides of the above loop, they would be identical, except for the pipe sizes, as the initial and final pressures are the same. Consequently, in any number of parallel lines of the same length, and joined together at the ends,

$$d = [d_1^{8/3} + d_2^{8/3} + d_3^{8/3} + \cdots d_n^{8/3}]^{3/8} \tag{95}$$

We can now write one equation with an equivalent diameter which will represent the sum of the separate equations for each branch of the loop, or parallel system

Using the above example, and substituting in equation 95,

$$d = [4.026^{8/3} + 2.81^{8/3}]^{3/8} = [41.02 + 15.72]^{3/8} = (56.74)^{3/8} = 4.55 \text{ in.}$$

We have now reduced the loop consisting of 4000 ft. of 3-in. on one branch and 2500 ft. of 4-in. on the other, to an equivalent single pipe 2500 ft. long, with an internal diameter of 4.55 in.

Complex pipe systems of different kinds may thus be simplified by means of equations 93, 94, and 95.

Comparative Pipe-Line Capacity. The comparative capacity of two pipes of different sizes, but with the same length and pressure drop, can be determined by writing equation 88 twice, obtaining

$$\frac{Q_1}{Q_2} = \frac{d_1^{8/3}}{d_2^{8/3}} = \left(\frac{d_1}{d_2}\right)^{8/3} \tag{96}$$

The same relationship also holds good for any number or combinations of pipes, in comparison with any other number or combinations of the same length, which may be expressed as

$$\frac{Q_1}{Q_2} = \frac{\Sigma_1\, d^{8/3}}{\Sigma_2\, d^{8/3}} \tag{97}$$

in which Σ_1 and Σ_2 are the summations of the eight-thirds powers of the pipes in systems 1 and 2, all pipes being of the same length, or being reduced to the same length and equivalent diameter by equation 94.

Alignment Charts. Problems involving pipe-line pressure drop are often conveniently solved by alignment charts, as extreme accuracy is generally not essential.

Chart 18 is for the solution of equation 88, for pipe-line pressure drop.

Charts 19 and 20 are used in connection with 18, to solve the pressure-drop radical $\sqrt{P_1^2 - P_2^2}$.

Chart 21 is designed for the solution of equations 93 and 94, for equivalent pipe length and diameter.

Chart 22 gives the equivalent size of loop lines for sizes over 6-in. It is based on equation 93.

Chart 23 solves equation 96 for the comparative capacities of pipes of equal length.

PROBLEMS

1. What is the capacity of an 8-in. standard pipe 2700 ft. long carrying gas of 0.78 gravity, if the initial pressure is 30 lb. gauge, and the pressure drop is to be held to 5 per cent? What will be the final pressure of the gas? *Solution:* Use Table XVIII. Length factor is 0.0192. Gravity factor, upon interpolation, is 55.23. Diameter function for 8-in. standard pipe is 262.1. Percentage pressure drop function for 5 per cent is 0.312. The capacity is $0.0192 \times 55.23 \times 262.1 \times (30 + 14.73) \times 0.312 = 3877$ M.c.f. The final pressure, from Table XIX, is 27.8 lb. per sq. in. gauge.

Use Table XXV. Factor for 2700 ft., by interpolation, is 88.6. Capacity is $88.6 \times (30 + 14.73) = 3963$ M.c.f. The larger capacity indicated by this method is due to the fact that the table is based on a gravity of 0.75, instead of 0.78.

Use Charts 18 and 19. First obtain the pressure-drop radical from Chart 19 by following the circle for 30 lb. gauge, on the left-hand side of the bottom half of the chart, to the right until it intersects the ordinate coming up from the bottom of the chart at 27.8 lb. final pressure, gauge. The value of the radical, as shown on the right-hand side, is 14. On Chart 18, draw a line from 14 on R to 0.78 on G, and get 2.43 on A. From 2.43 on A draw a line to 2700 ft. on L, and get 3.43 on B. From 3.43 on B draw a line to 8-in. standard pipe on D, and get 3990 M.c.f. on Q for the capacity of the line.

2. What is the capacity of 5000 ft. of $2\frac{1}{2}$-in. line if the gravity of the gas is 0.70 and the initial and final pressures are 200 and 150 lb. gauge? *Solution:* On Chart 19 secure the value of the pressure-drop radical by noting the intersection of the 150-lb. vertical line with the 200-lb. initial pressure circle and projecting to the right, obtaining a value of 135 for R. On Chart 18, draw a line from 13.5 on R to 0.70 on G, and get 2.5 on A. From 2.5 on A draw a line to 5000 on L, and get 2.95 on B. From 2.95 on B draw a line to $2\frac{1}{2}$ on the right-hand side of D, and get 125,000 standard cubic feet on Q. As 13.5 was used on the R scale instead of 135, multiply the capacity figure by 10, obtaining a capacity of 1,250,000 standard cubic feet per 24 hours.

3. An oil well produces 1,000,000 cu. ft. of gas per day, which is drawn off by a compressor through 20,000 ft. of 8-in. pipe. The gravity of the gas is 0.80. What pressure must be maintained at the compressor to produce atmospheric pressure at the well? *Solution:* On Chart 18, draw a line from 8-in. on the right side of D through 1000 M.c.f. on Q, and get 1.15 on B. From 20,000 on L draw a line through 1.15 on B, and get 1.25 on A. From 0.80 on G draw a line through 1.25 on A, and get 10 on R. Using the bottom half of Chart 19, start at 10 on the right-hand side and follow this line until it intersects the zero circle of initial pressure. Project this point downward, and get 8 in. vacuum for the pressure at the compressor.

4. What size of pipe is required to transport 25,000,000 cu. ft. of 0.75 gravity gas per 24 hours a distance of 4 miles, if the initial and final pressures are to be 20 and 5 lb. gauge? *Solution:* Obtain the pressure-drop radical from the lower half of Chart 19, by noting the intersection of the 5-lb. vertical line with the 20-lb. circle. Project this point to the right, and get 28.5 as the value of R. On Chart 19 draw a line from 28.5 on R to 0.75 on G, and get 4.9 on A. From 4.9 on A draw a line to 21,120 ft. on L, and get 2.9 on B. From 2.9 on B draw a line through 25,000 M.c.f. on Q, and get 18 in. on D as the internal diameter of the pipe required.

5. How much 6-in. pipe will have the same pressure drop as 800 ft. of 4-in. pipe? *Solution:* On Chart 21 draw a line from 4 in. on the left side of the left-hand scale to 800 on the right-hand scale, and get 5.15 on the center scale. Draw a line from 6 in. on the left of the left-hand scale through 5.15 on the center scale, and get 6750 on the right-hand scale, which is the length of 6-in. pipe that will give the same pressure drop as 800 ft. of 4-in. pipe.

6. What size pipe, 3000 ft. long, will have the same pressure drop as 575 ft. of 4-in. standard pipe? *Solution:* On Chart 21 draw a line from 4 in. on the left side of the left scale to 575 on the right-hand scale, and get 5.4 on the center scale. From 3000 on the right-hand scale draw a line through 5.4 on the center scale, and get a pipe of 5.5-in. internal diameter on the left-hand scale.

7. An 8-in. and a 12-in. line parallel each other and are of the same length. What size pipe would have the same capacity as both of them if they were connected in parallel at both ends? *Solution:* On Chart 22, draw a line from 8 in. on the left-hand scale to 12 in. on the right-hand scale, and get approximately 13.45 on the center scale, as the size of the equivalent line with the same capacity of both. Check by using Table XVIII. Factors for 8-in. and 12-in. lines are 262.1 and 769.97. Adding them, we get 1032.07. Entering the table again, we find 1033 for a 13.5-in. line.

8. A 12-in. gas line laid on the bed of a river has been washed out. How many 6-in. lines must be laid to take its place? *Solution:* Factors for 12-in. and 6-in. lines from Table XVIII are 769.97 and 122.33. The ratio of the factors or 769.97/122.33 = 6.3. We must therefore install seven 6-in. lines to replace the 12-in. line which was washed out.

9. A piece of 12-in. pipe is connected between two lengths of 16-in. pipe. What size pipe should be connected in parallel with the 12-in. pipe to bring the capacity of this section of the line up to that of the 16-in.? *Solution:* On

Chart 22 draw a line from 12 in. on the left-hand scale through 15.25 in. on the center scale, and get 11.5 in. for the internal diameter of the required pipe on the right-hand scale. Check by Table XVIII. Factor for 16-in. pipe minus factor for 12-in. pipe is 1430.16 − 769.97 = 660.19, which is the factor for a pipe approximately 11.41 in. in diameter.

10. What is the comparative capacity of an 8-in. and a 10-in. line? *Solution:* On Chart 23, draw a line from 8 in. on the right-hand scale to 10 in. on the middle scale, and get 1.7 on the left-hand scale. The ratio of capacities is therefore 1.7. Using Table XVIII for more accurate results, 488.31/262.1 = 1.863.

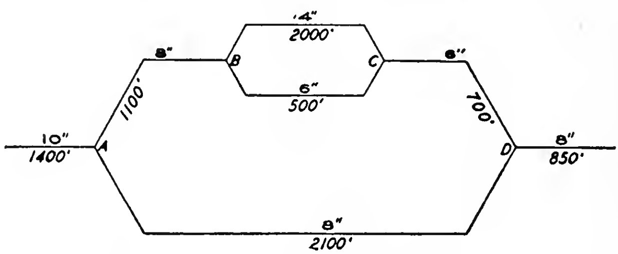


FIG. 32.

11. Reduce the pipe system shown in Fig. 32 to an equivalent length of 8-in. pipe. *Solution:* First reduce the loop from *B* to *C* to two branches of the same length by referring to Chart 21. The result is a loop 2000 ft. long with 4-in. on one side and 7.8-in.-inside-diameter pipe on the other, which is equivalent to 2000 ft. of 8.27-in.-diameter pipe. Next reduce the upper part of the system from *A* to *D* to 6-in. pipe. The 1100 ft. of 8-in. is equivalent to 240 ft. of 6-in. The 2000 ft. of 8.27-in. pipe is equivalent to 390 ft. of 6-in. pipe. The value of the upper part of the system from *A* to *D* is therefore 240 + 390 + 700 = 1330 ft. of 6-in. This is equivalent to 2100 ft. of 6.6-in.-internal-diameter pipe. We now have a loop 2100 ft. long of 8-in. and 6.6-in. pipe, which is equivalent to a single pipe 9.6 in. in inside diameter. The 9.6-in. pipe is equivalent to 830 ft. of 8-in. The 1400 ft. of 10-in. is equivalent to 470 ft. of 8-in. The total value of the system is therefore 470 + 830 + 850 = 2150 ft. of 8-in. pipe.

The above problem may be solved by combining the different parts in any manner desired, but the result should be the same, no matter in what order the work is done.

12. A compressor plant and gas-gathering system are shown in Fig. 33. The production of the wells is given in thousands of cubic feet per 24 hours. The average gravity of the gas is 0.75. If the compressors maintain a vacuum of 10 in. of mercury at the plant intake, what will be the pressure on each of the wells? *Solution:* The total gas handled by the compressors is 2000 + 1500 + 3000 = 6500 M.c.f. per 24 hours. First compute the line drop from the compressors to point *A*, using Charts 18 and 19. The pressure at *A* is found to be 4 in. vacuum. The flow from *A* to well 1 is 2000 M.c.f., and the pressure is found to be ½ lb. above atmosphere. From *A* to *B*, the flow is 4500 M.c.f., and the pressure at

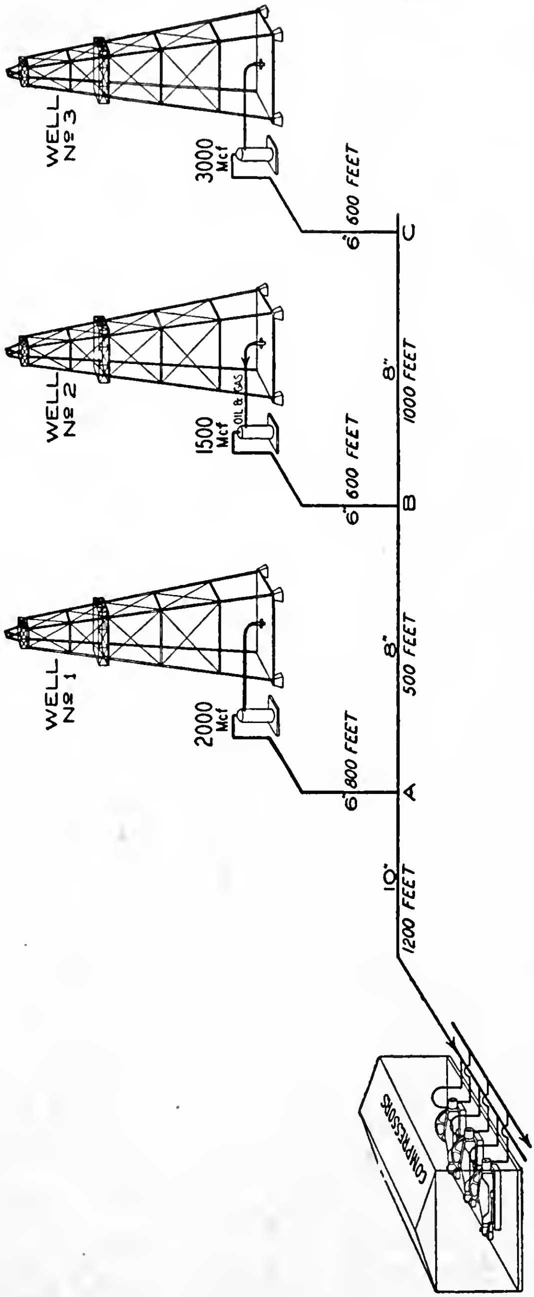

WELL
No 1
2000
Mcf
WELL
No 2
1500
Mcf
OIL & GAS
WELL
No 3
3000
Mcf
6" 800 FEET
A
8" 500 FEET
B
6" 600 FEET
8" 1000 FEET
C
6" 600 FEET
10" 1200 FEET
COMPRESSORS

Fig. 33.

B is approximately atmosphere. Other pressures are: well 2, 1 lb. gauge; point C, $1\frac{1}{4}$ lb. gauge; well 3, $4\frac{1}{2}$ lb. gauge. The same result for the pressure at well 3 may be obtained by reducing the 1000 ft. of 8-in. and the 600 ft. of 6-in. pipe to an equivalent length of 3750 ft. of 8-in. by means of Chart 21. In using Chart 18 in this problem, it will be found that the values often run off the scale as the work progresses. In such cases, simply take 10 times as much gas as is called for, and the result will be that the value of the pressure drop radical will be 10 times too much. Divide the resulting value of the radical by 10 and use in Chart 19. It is well to note that the radical and Q are the only variables on Chart 18 that may be so multiplied or divided by 10 without affecting the results.

If it is desired to carry lower pressures on the wells, it will be necessary to assume a lower vacuum at the compressors and recompute the problem, or the pipe sizes may be increased, if possible. As pipes are seldom available in fractional sizes, it will be evident that it is impossible to design a system with exactly the same pressure on all wells, and even if such a design were possible, a falling off of production on one of the wells would spoil the arrangement. The best method seems to be to put in as large pipes as possible, considering the cost, and after assuming a pressure at the compressors, compute the pressure drop throughout the system as given above. If the pressures at the wells are too high or too low, then assume another pressure at the compressors and solve through again until the desired result is obtained.

CHAPTER XVIII

GAS MEASUREMENT BY ORIFICE METER

Accurate compressor calculations are impossible without a consistent and reasonably accurate method of gas measurement. This chapter is merely an introduction to the study of the metering of gas by commercial methods. An exhaustive consideration of the subject would be beyond the scope of this book.

Small quantities of gas may be measured by displacement or volumetric meters, a common example of which is the domestic gas meter. This type records the actual volume passing through it but takes no account of the pressure. It is satisfactory for domestic gas use, where the variations in pressure are small. For large variations in pressure,

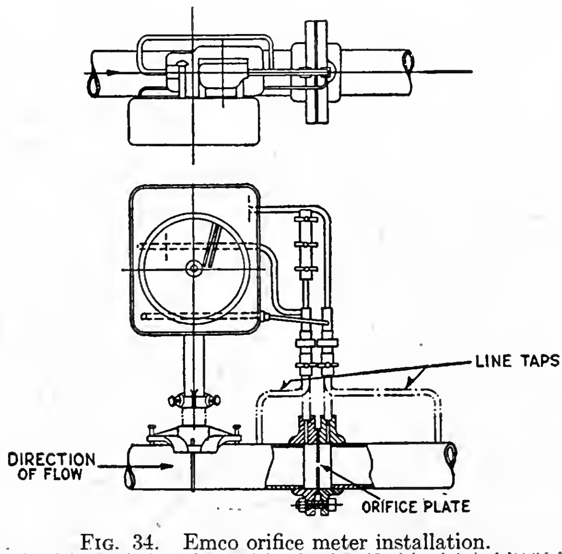


Fig. 34. Emco orifice meter installation.

the meter must be equipped with a recording pressure device, or integrator of some sort, or its readings are useless. A volumetric meter which could handle large volumes of gas would be unwieldly in size.

For commercial purposes, the orifice meter, with recording gauge and differential indicator that trace a record on a circular chart, seems to

give satisfaction under varying conditions. The Venturi meter is suitable for large quantities of gas with a low pressure drop but lacks flexibility. The calibrated nozzle, in various forms, is often used for special tests.

Figure 34 represents an orifice meter, with Emco differential gauge and recorder. The essential parts of the meter consist of a plate about

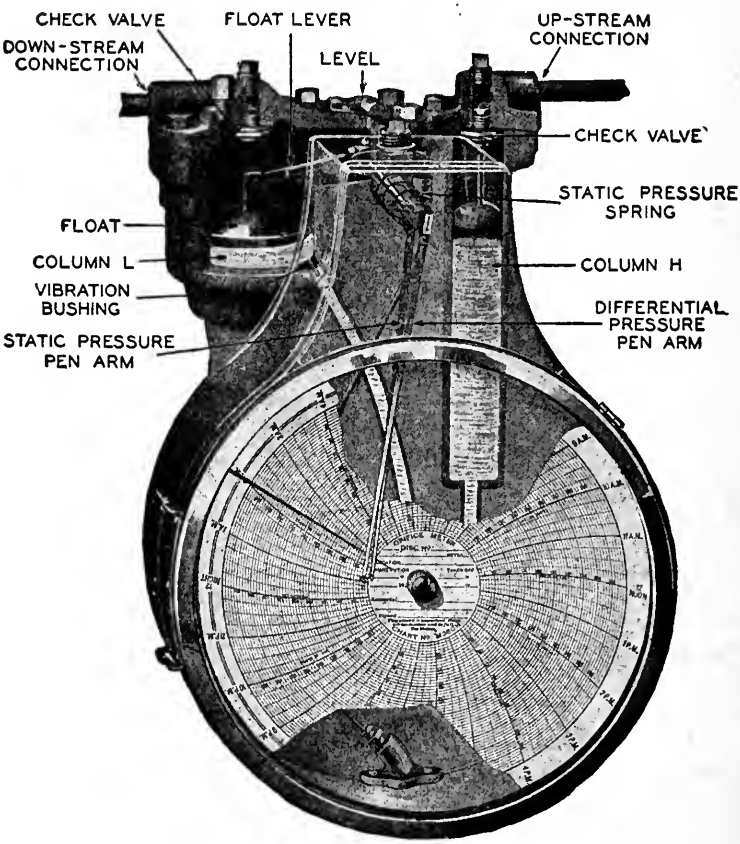


FIG. 35. Westcott orifice meter.

$\frac{1}{4}$ in. thick, having a round orifice or hole, considerably smaller in diameter than the pipe in which it is placed. Pressures are secured either a short distance, usually about 1 in., on either side of the orifice plate (flange taps), or $2\frac{1}{2}$ pipe diameters upstream and 8 diameters downstream from the plate (line taps). Various methods of locating the line taps have been proposed, but the $2\frac{1}{2}$- and 8-diameter arrangement is the most common.

The pressure drop across the orifice when gas is flowing, usually known as the "differential," is measured in inches of water by a mercury manometer which supports an iron float on one branch, to operate the recording mechanism. A sectionized view of a Westcott orifice meter is shown in Fig. 35.

The static pressure is taken upstream in line taps and downstream in flange taps, on account of the difference in coefficients used in calculations. It is well to note here, however, that, though the above statement is true for the abbreviated method of calculation given here, it does not apply to the latest data now in use. The introduction of a new expansion coefficient makes it possible to use downstream pressures in all cases.

The general orifice meter formula, which is based on the Chezy equation (equation 87), is largely empirical in nature, and consequently its derivation will not be attempted here. After the insertion of constants to reduce to standard conditions of 14.73 lbs. per sq. in. abs. and 60°F., the expression becomes

$$Q = 338.2\, C_v\, d^2 \sqrt{\frac{hP}{G}} \tag{98}$$

where Q is given in standard cubic feet per hour; d is the diameter of the orifice in inches; h is the differential pressure drop across the orifice in inches of water; P is the static pressure in pounds per square inch absolute; G is the specific gravity of the gas, based on air = 1.00. C_v is the so-called efficiency coefficient of the orifice, and is taken as

$$C_v = 0.606 + 1.25\,(X - 0.41)^2 \tag{99}$$

for flange connections, and

$$C_v = 0.58925 + 0.2725\,X - 0.825\,X^2 + 1.75\,X^3 \tag{100}$$

for $2\frac{1}{2}$- and 8-diameter line connections. In the two above equations, X is the ratio of the diameter of the orifice to the internal diameter of the pipe.

In equation 98 the temperature of the gas is taken as 60°F. For other temperatures, multiply by the factor

$$C_t = \sqrt{\frac{520}{460 + T}} \tag{101}$$

The orifice meter record is made upon a circular chart, upon which pens actuated by pressure gauge and differential manometer inscribe a continuous record of the pressure and differential readings needed for volume calculations. When the graduations on the paper chart are

uniformly spaced, the differential and gauge pressures may be read directly and used in the flow formula.

In order to avoid the necessity of a large variety of charts, graduated to suit the requirements of a number of different meters, it has become popular to use the so-called "square-root" chart, the graduations of which are proportional to the square root of the differential and absolute pressure. The markings read from 0 to 10. It is impossible to read off the pressure and differential from these charts without special tables.

The relation between uniform and square-root charts may be expressed as

$$\sqrt{h\,P} = h_s\,P_s\,0.01\sqrt{R_h\,R_p} \tag{102}$$

where h and P are the differential and pressure, ordinarily read from the uniform charts; h_s and P_s are the differential and pressure readings from the square-root charts; and R_h and R_p are the differential and pressure ranges of the meter. Thus, if the range of the differential gauge is 20 in. of water, and the range of the pressure spring is from 0 to 50 lb., then

$$\sqrt{h\,P} = h_s\,P_s\,0.01\sqrt{50 \times 20} = 0.3162\,h_s\,P_s$$

For the commercial measurement of gas in large quantities, Reynolds' number, expansion, superexpansibility, and other correction factors are used, which are described in publications of the orifice meter manufacturers and various technical associations.[1]

To provide a rough check on orifice meter calculations, the following alignment charts have been furnished: Chart 24, giving values of $\sqrt{hP}$, the pressure extension; Chart 25, orifice meter flow for flange connections; Chart 26, flow with $2\frac{1}{2}$- and 8-diameter line connections; and Chart 27, for recommended orifice sizes.

[1] See "Tentative Standard Procedure for the Measurement of Natural Gas with Orifice Meters," the report of a joint committee of the California Natural Gasoline Association, the Pacific Coast Gas Association, the Southern California Meter Association; and based on the American Gas Association Gas Measurement Committee Report 2, which is issued as *Bulletin* TS-353, by the California Natural Gasoline Association, Los Angeles, 1935.

The Orifice Meter, published by the Pittsburgh Equitable Meter Co., Pittsburgh, Pennsylvania.

Orifice Meter Constants, Handbook E-2, published by the American Meter Co., Erie, Penn. (now reprinted). See also *Handbook E-2R.*

For a description of the various phases of gas measurement and of different apparatus for determining specific gravity, etc., see *Natural Gas* by Lester C. Lichty, John Wiley & Sons; "Discharge Coefficients of Square-Edged Orifices for Measuring the Flow of Air," *Research Paper* 49, Bureau of Standards; and "Gas-Measuring Instruments," Bureau of Standards *Circular* 309.

Chart 27 is based on the formula

$$Q_d = \frac{24\ C_0}{1000}\sqrt{\frac{Ph}{2}} \tag{103}$$

where Q_d is the desirable flow in thousands of standard cubic feet per 24 hours (M.c.f.); C_0 is the hourly air coefficient for an orifice of approximately one-half the internal diameter of the pipe; P is the static pressure at which the meter is to operate; and h is the maximum differential of the meter in inches of water. The formula gives about 35 per cent of maximum flow possible with the same pipe size and with the largest recommended orifice size. It is designed to place the differential pen in the center of the chart, to allow for possible variations of flow.

PROBLEMS

1. How much gas in standard cubic feet per 24 hours will pass through a 3-in. orifice in a 6-in. pipe line if the static pressure is 56 lb. per sq. in. gauge and the drop across the orifice is 22 in. of water? Line connections $2\frac{1}{2}$ and 8 diameters are used, and the gravity of the gas is 0.68. *Solution:* First secure the pressure extension from Chart 24 by drawing a line from 22 on the center scale to 56 on the left-hand scale, obtaining 39 on the second scale. Then on Chart 26 draw a line from 39 on E to 0.68 on G, obtaining 6.05 on the turning scale A. Between scales Q and G on the chart will be found a number of vertical lines representing pipe sizes, intersected by curved lines representing orifice sizes. Follow the 6-in. vertical pipe size line until it crosses the curved line marked 3-in., and project this point to the left with a small triangle until it intersects the 16-in. line at right angles, at a point corresponding to an hourly coefficient of 2250. Next draw a line from 6.05 on A to 2250 on C, and obtain 2500 on Q. The flow will therefore be 2,500,000 standard cubic feet per 24 hours.

2. It is required to pass 1,000,000 cu. ft. of gas per 24 hours through a 6-in. pipe having a flange-connected orifice; gravity of gas is 0.80; static pressure is 29 lb. gauge, and the average differential is 25 in. of water. What size orifice should be used? *Solution:* Obtain pressure extension from Chart 24 by drawing a line from 25 on the center to 29 on the first scale, obtaining 33 on the second scale. On Chart 25 draw a line from 33 on E to 0.80 on G, cutting A at 5.2. From 5.2 on A draw a line through 1000 M.c.f. on Q, which will pass through 1150 on scale C. Project this value to the right at right angles until it strikes the vertical line for a 6-in. pipe. The curved line marked $2\frac{3}{8}$ in. passes through the 6-in. line at this point and is the size of orifice required.

3. Required, the most desirable line size for 4500 standard M.c.f. per day of 0.64 gravity gas at 80°F. and 200 lb. gauge pressure, using a 50-in. differential orifice meter. *Solution:* Secure the corresponding air rate by dividing the desired flow by the gravity correction factor $\sqrt{1.00/0.64}$, and by the temperature correction factor $\sqrt{520/540}$. The resulting air rate is 3669 M.c.f. On Chart 27

draw a line from 200 lb. gauge on the left-hand scale to 3669 on the right-hand scale, and get a point between the 6-in. and the 8-in. pipes on the center scale, using the left-hand side. Use the 6-in. line.

4. Required, approximate maximum flow rate for the gas in problem 3, using a 6-in. pipe and 50-in. meter. *Solution:* In Chart 27, draw a line from 200 lb. on the left-hand scale through 6 in. on the left side of the center scale, and get an air rate of 3300 M.c.f. on the right-hand scale. The approximate maximum flow for a 6-in. pipe with largest permissible orifice will be $[3300/0.35] \times \sqrt{1.00/0.64} \times \sqrt{520/540} = 11{,}570$ M.c.f. per 24 hours of the 0.64 gravity gas of problem 3. It is evident that the 6-in. pipe size chosen will be adequate for the 4500 M.c.f. desired.

CHAPTER XIX

USE OF ALIGNMENT CHARTS

An alignment chart is nothing more or less than the graphical representation of a mathematical formula, arranged in such a way that any variable can be solved for if the others are known. The simplest form of a chart consists of three scales so arranged that a straight line drawn from a known value on one scale to a known value on another will pass through the third scale at a value which will satisfy a given equation.

Take, for example, a simple multiplication and division chart such as Fig. 36. Let Scale A represent the multiplier, B the product, and C the multiplicand. To multiply 3 by 4, draw a line from 3 on A to 4 on C, and obtain the result 12 on B.

Suppose, however, that A and B are known and C is desired, which would amount to the problem of what number multiplied by A would give B, or what would be the quotient of B divided by A. If A is 6 and B is 24, draw a line through these two values as before, and it will pass through 4 on Scale C.

It is sometimes desirable to find other sets of values which will give the same result in a certain equation as a known pair of values. Thus, if it is known that $AC = B$, or $9 \times 4 = 36$, we might want to know how the value of C would be affected if A were reduced in value from 9 to 6, with the same resulting value of the product B. In this case, it is only necessary to pivot the straight edge around the value of 36 on Scale B, and any set of values intersected on A and C will satisfy the equation.

For charts containing more than three variables, it is necessary to work in sets of three scales arranged in interlocking groups. For example, such a chart as Fig. 37 may bear the key

$$A\ B\ F$$
$$B\ C\ K$$
$$C\ Q\ G$$

in which B and C are merely dummy or turning scales, usually graduated with marks a uniform distance apart, and numbered from 0 to 10. These graduations make it unnecessary to draw lines on the chart, as

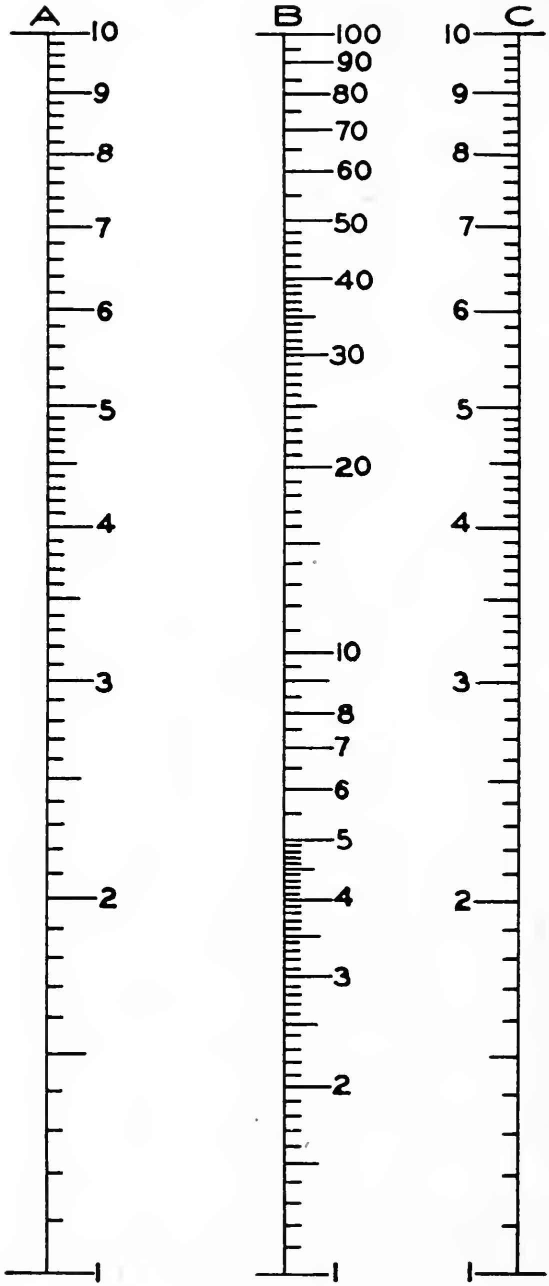

A
10
9
8
7
6
5
4
3
2
1
B
100
90
80
70
60
50
40
30
20
10
8
7
6
5
4
3
2
1
C
10
9
8
7
6
5
4
3
2
1

Fig. 36.

the scale readings may be noted instead, and used for further intersections. If the chart in Fig. 37 is to be solved in the order given in the key, the first step would be to apply a straight-edge to the known values on A and F, and a value will be obtained on B, which should be carefully noted or preferably jotted down on a piece of scratch paper. As B is now known, the triad BCK may be solved by going from B to K and obtaining the point where the line crosses C. The final result Q is obtained by going from C to G.

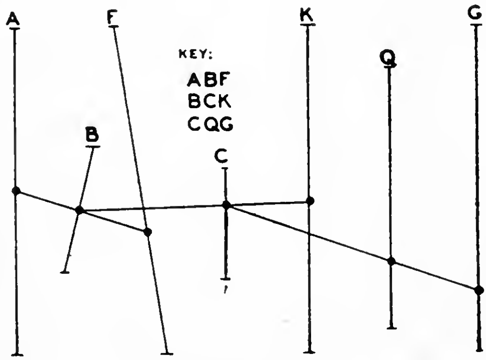


Fig. 37.

It is possible, however, to obtain any one of the values A, F, K, Q, or G if all the others are known. If the value of K is not known, it may be found by solving ABF for B and CQG for C, and then drawing a line through B and C and getting the required intersection on K. Other solutions may be made in a similar manner if it is remembered that all scales are to be used in groups of three, *and only as given in the key on the chart.* Under no circumstances, for example, would it be permissible to join A and K, or G and F, on the chart in Fig. 37.

The most satisfactory straight-edge for use with alignment charts is a piece of heavy celluloid with a straight line scratched on the under side and filled with ink.

For general information on the construction of alignment charts, see:
Graphical and Mechanical Computation, by Lipka, John Wiley & Sons, 1918.
The Construction of Graphical Charts, by Peddle, McGraw-Hill Book Co., 1919.
The Construction of Alignment Charts, by Swett, John Wiley & Sons, 1928.

APPENDIX

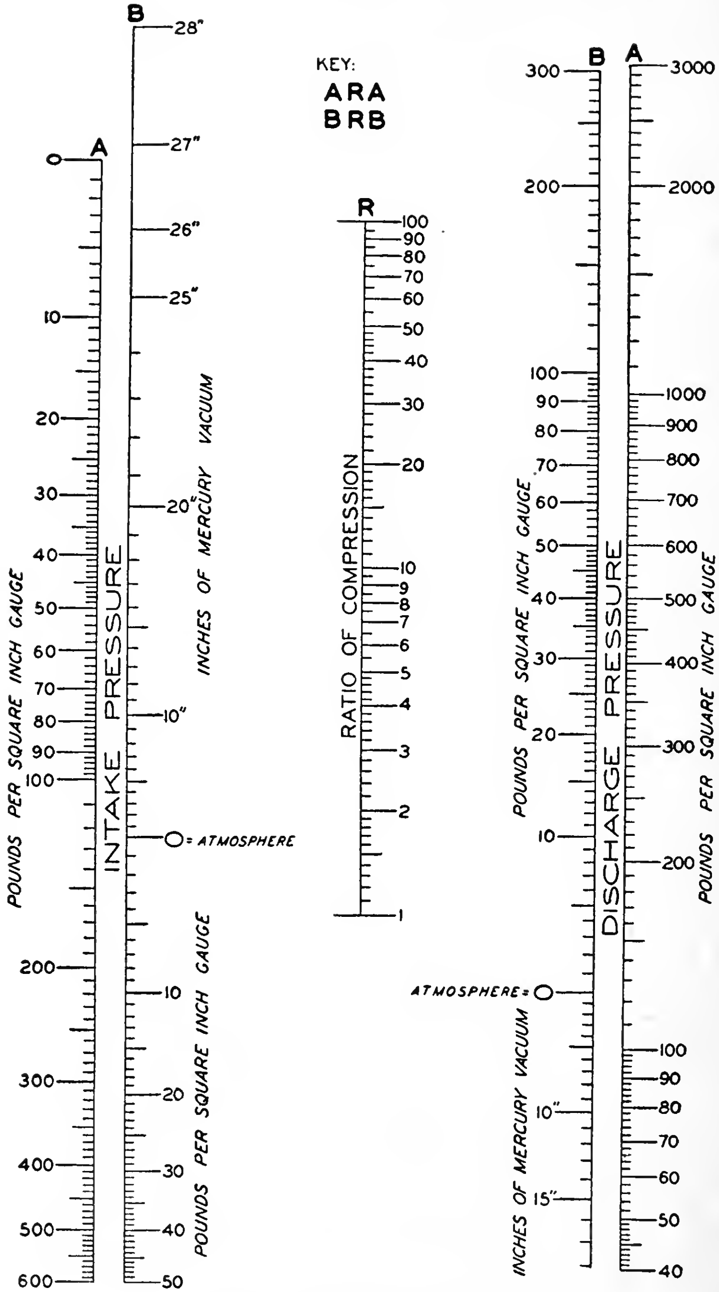


CHART 1. Ratio of Compression.
See problem 1, Chapter XII.

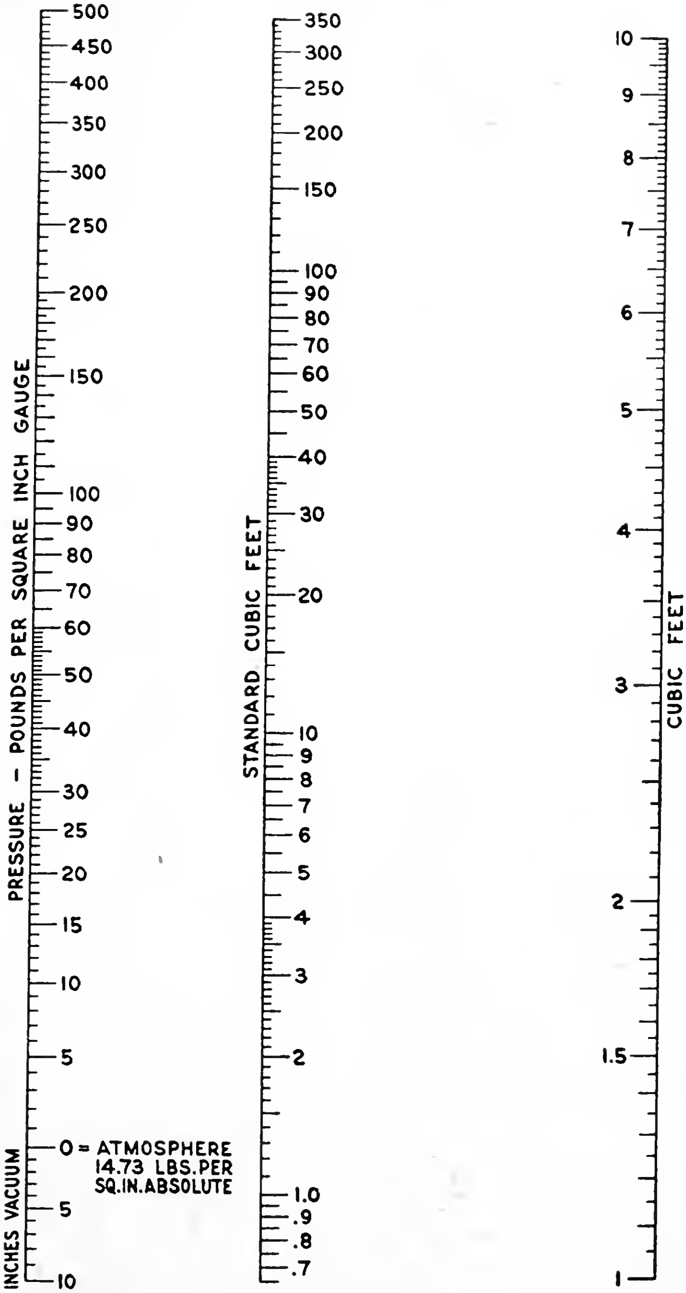


CHART 2. Gas Volume Correction.
See problem 2 and 3, Chapter XII.

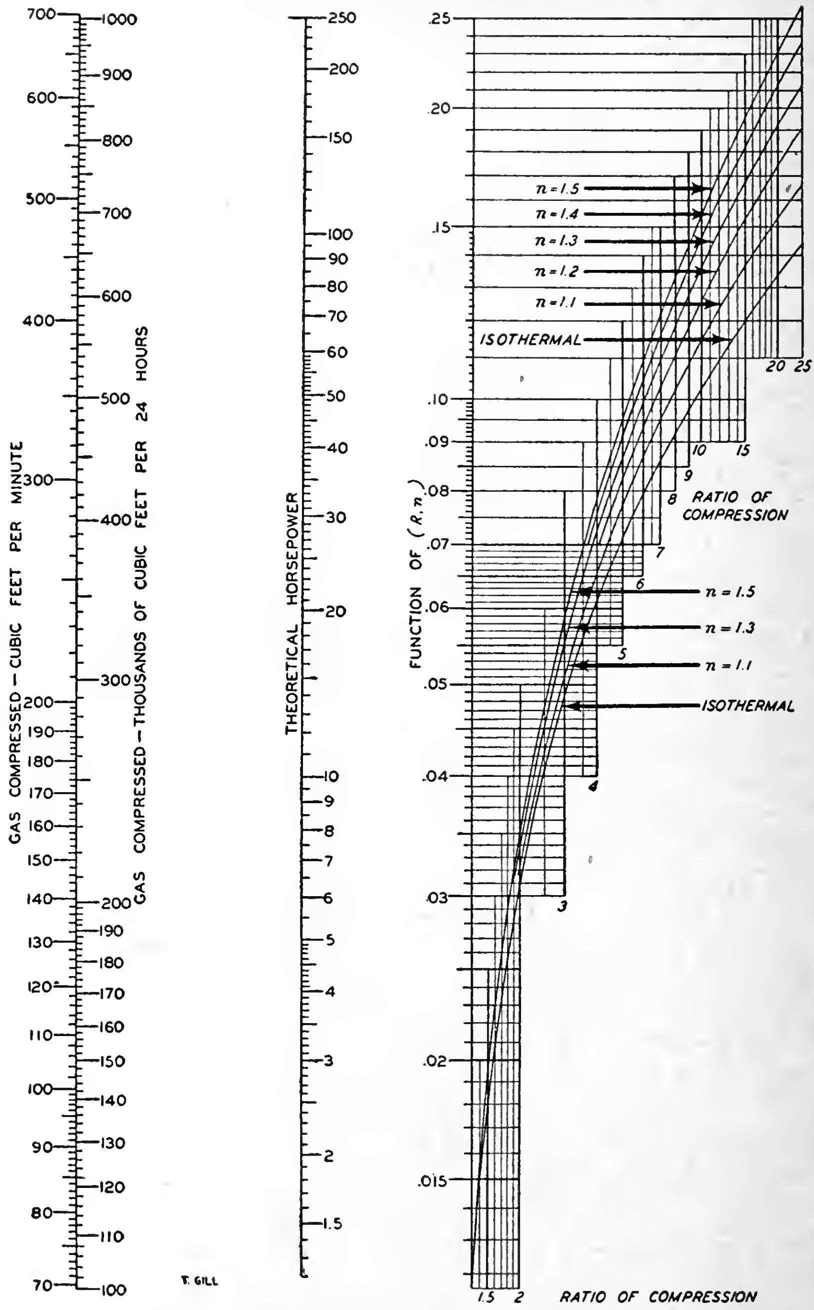


CHART 3. Theoretical Horsepower of Compression.
Based on equations 11, 12, 16, and 17. See problems 5 and 6, Chapter XII.

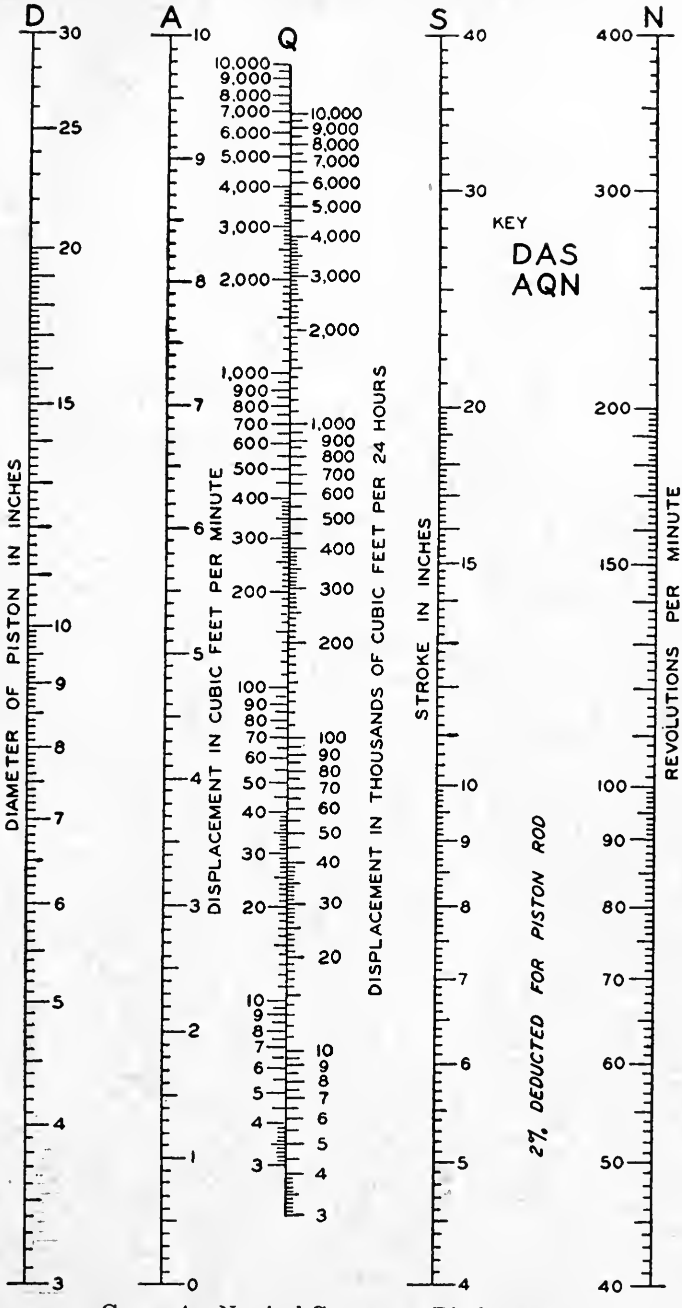


CHART 4. Nominal Compressor Displacement.
Based on equations 19 and 20. See problems 8 and 9, Chapter XII.

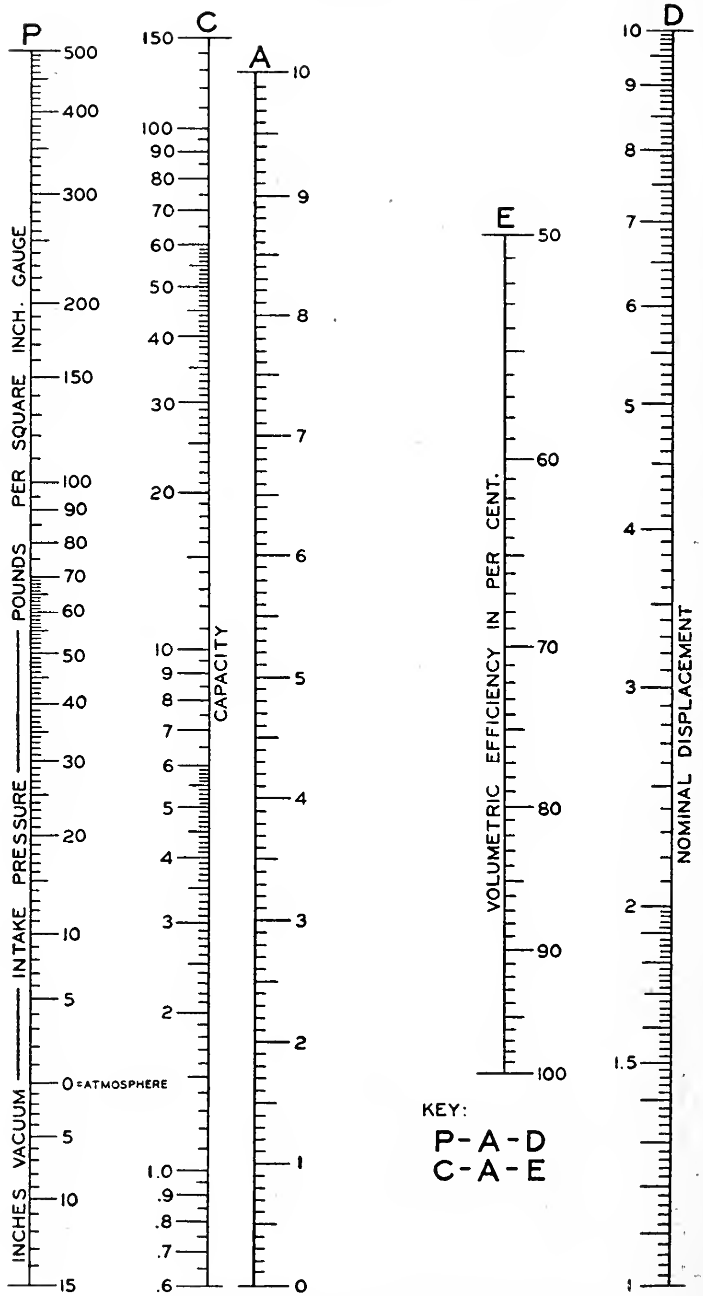


Chart 5. Compressor Capacity and Displacement.
See problems 10 and 11, Chapter XII.

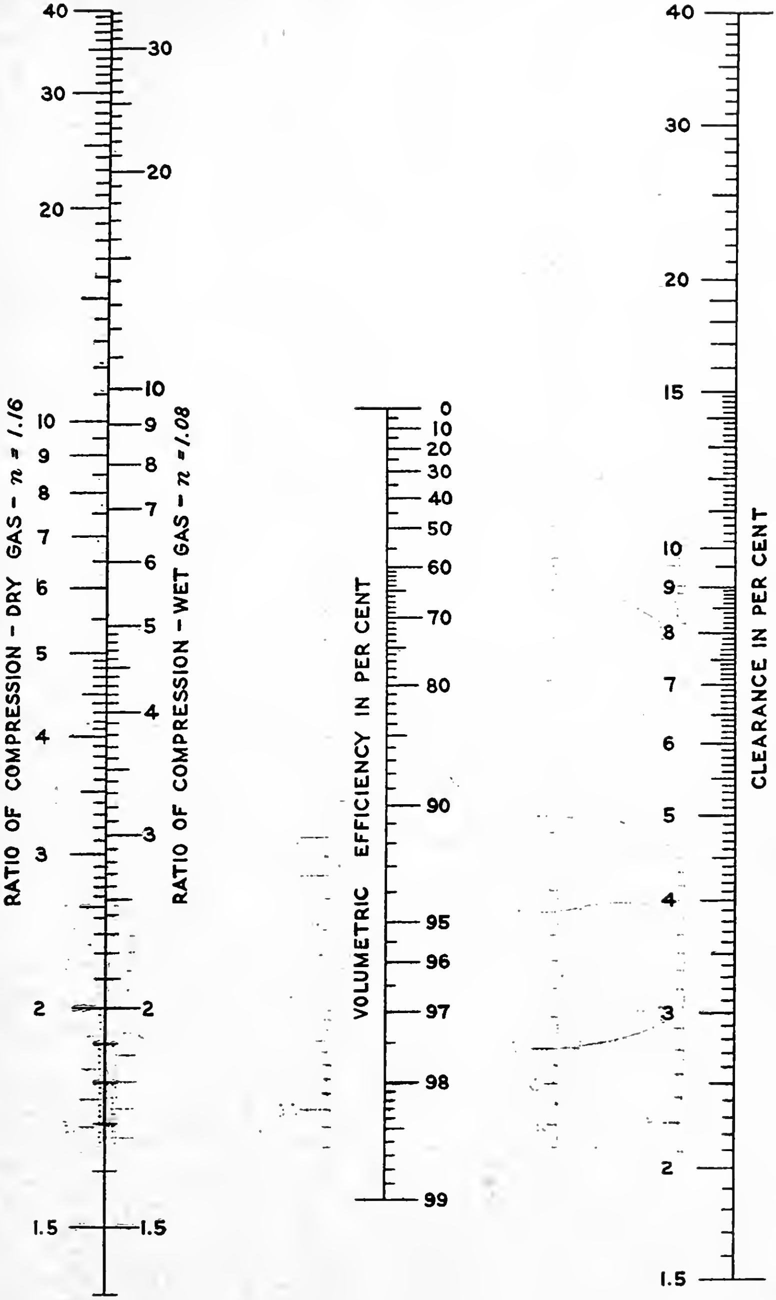


CHART 6. Volumetric Efficiency.
Based on equation 23. See problem 14, Chapter XII.

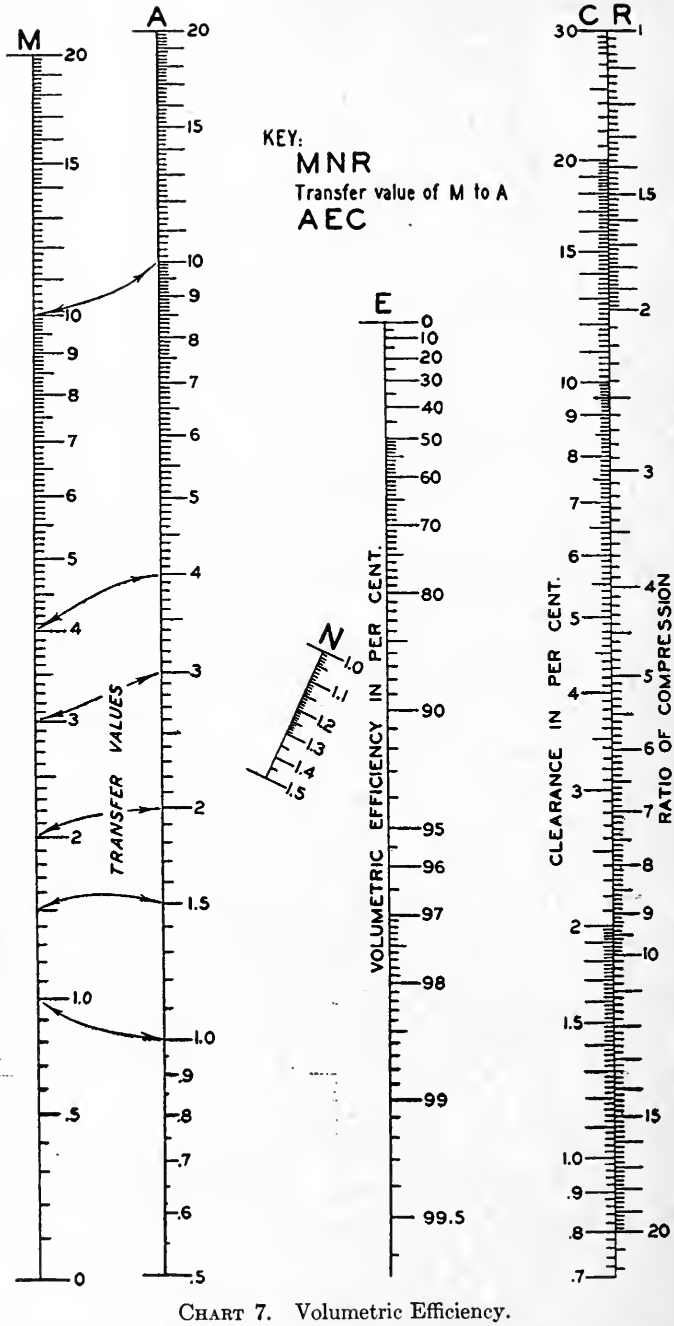


CHART 7. Volumetric Efficiency.
Based on equation 23. See problems 15 and 16, Chapter XII.

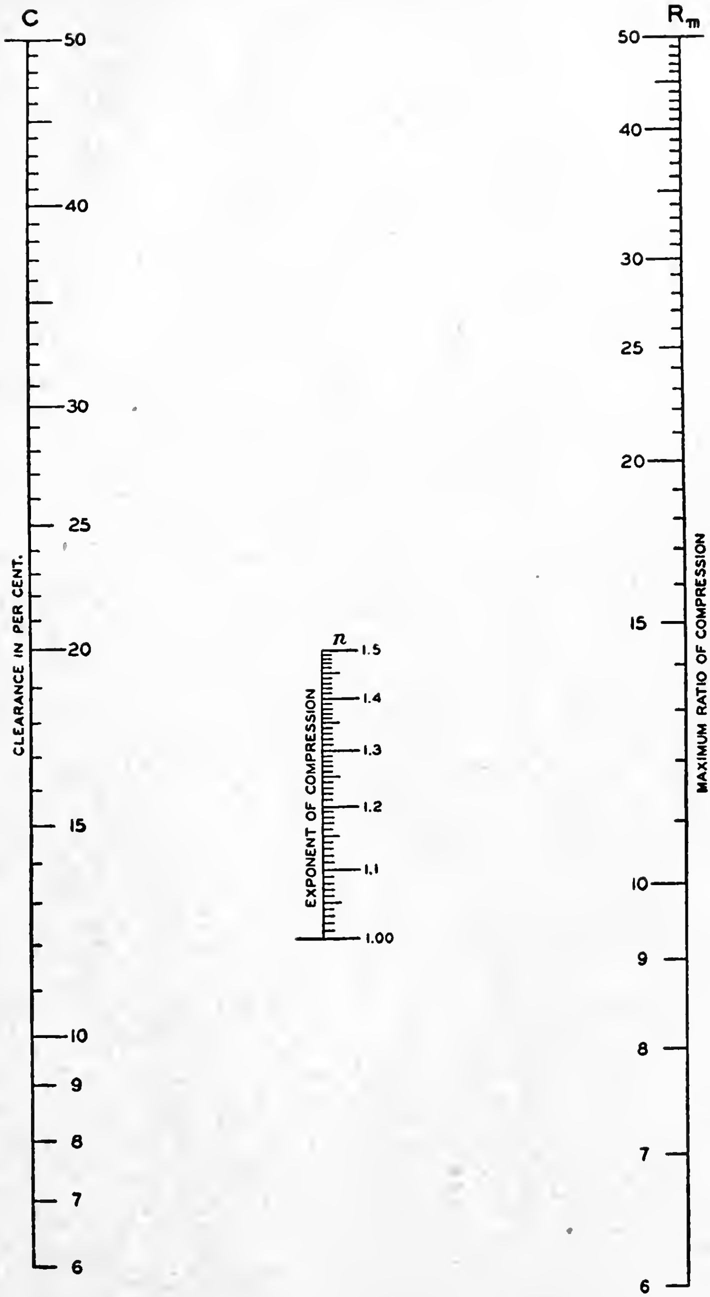


CHART 8. Maximum Ratio of Compression.
Based on equation 24. See problems 18 and 19, Chapter XII.

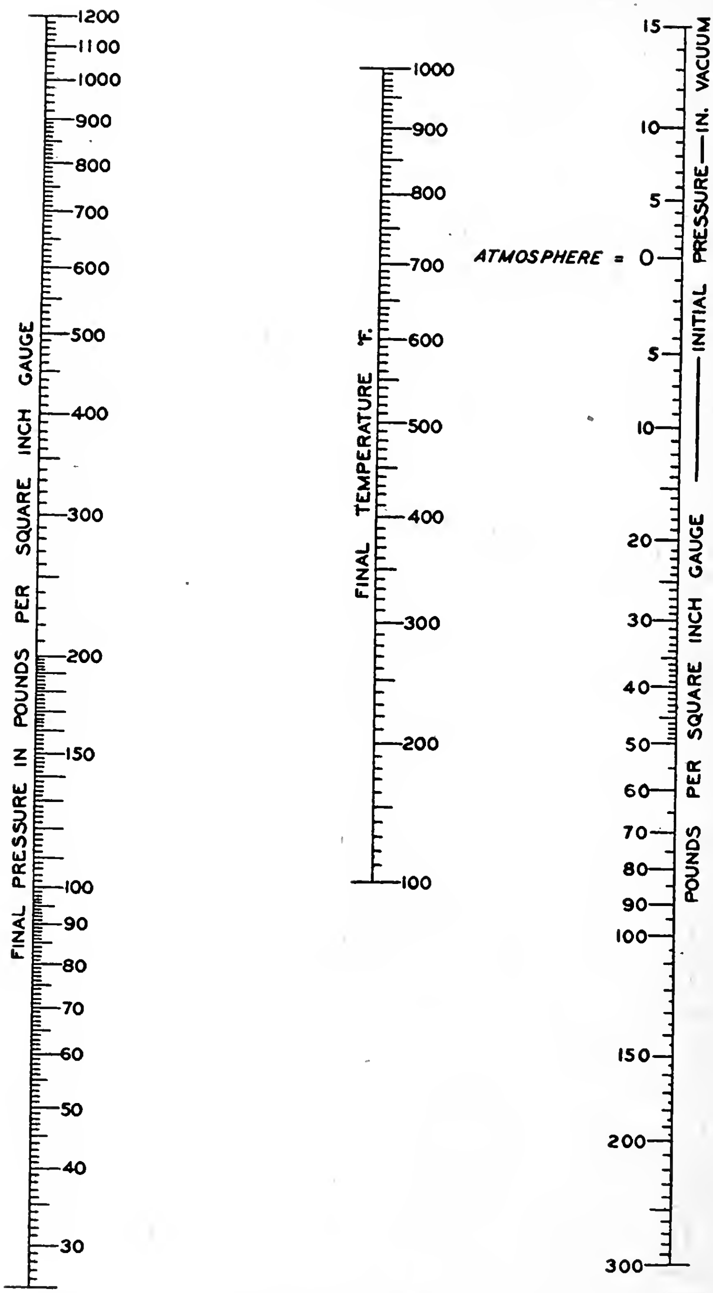


CHART 9. Temperature Rise of Gas During Compression.
Based on equation 29, with initial temperature of 80° F. and n = 1.26
See problem 20, Chapter XII.

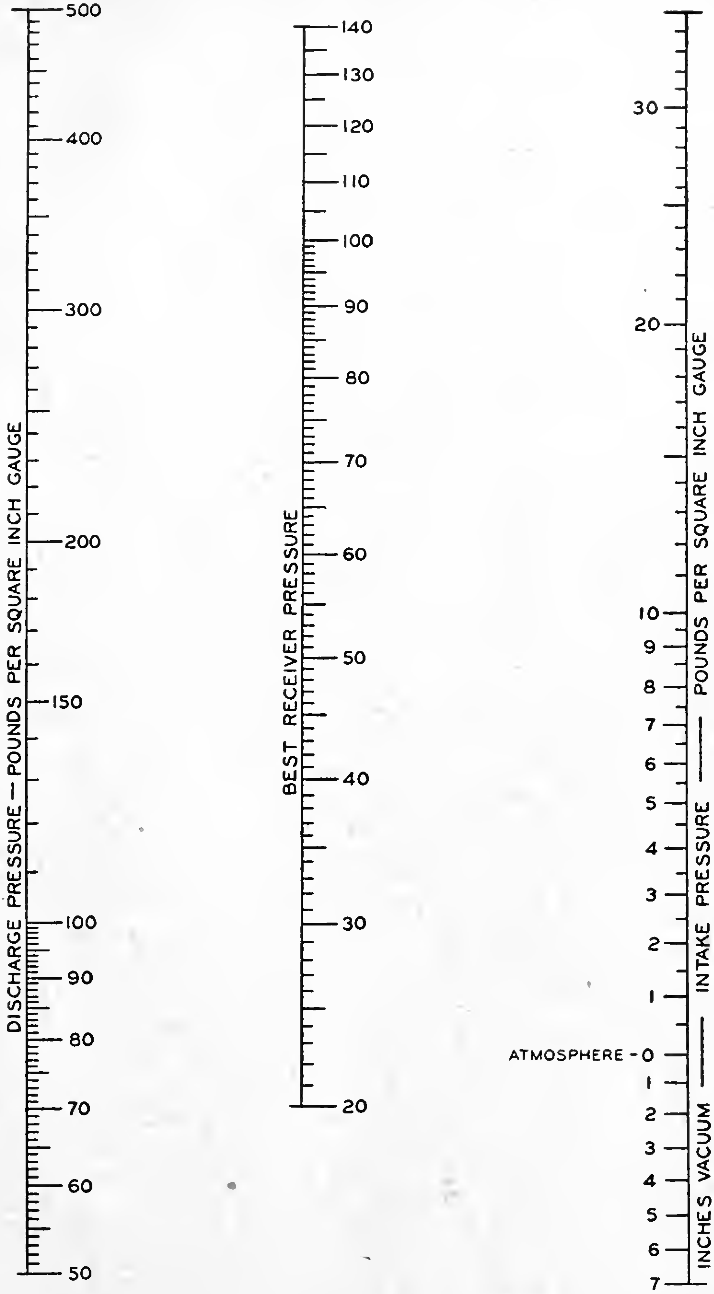


Chart 10. Intercooler Pressure for Two-Stage Compression: *Based on equation 30. See problem 21, Chapter XII.*

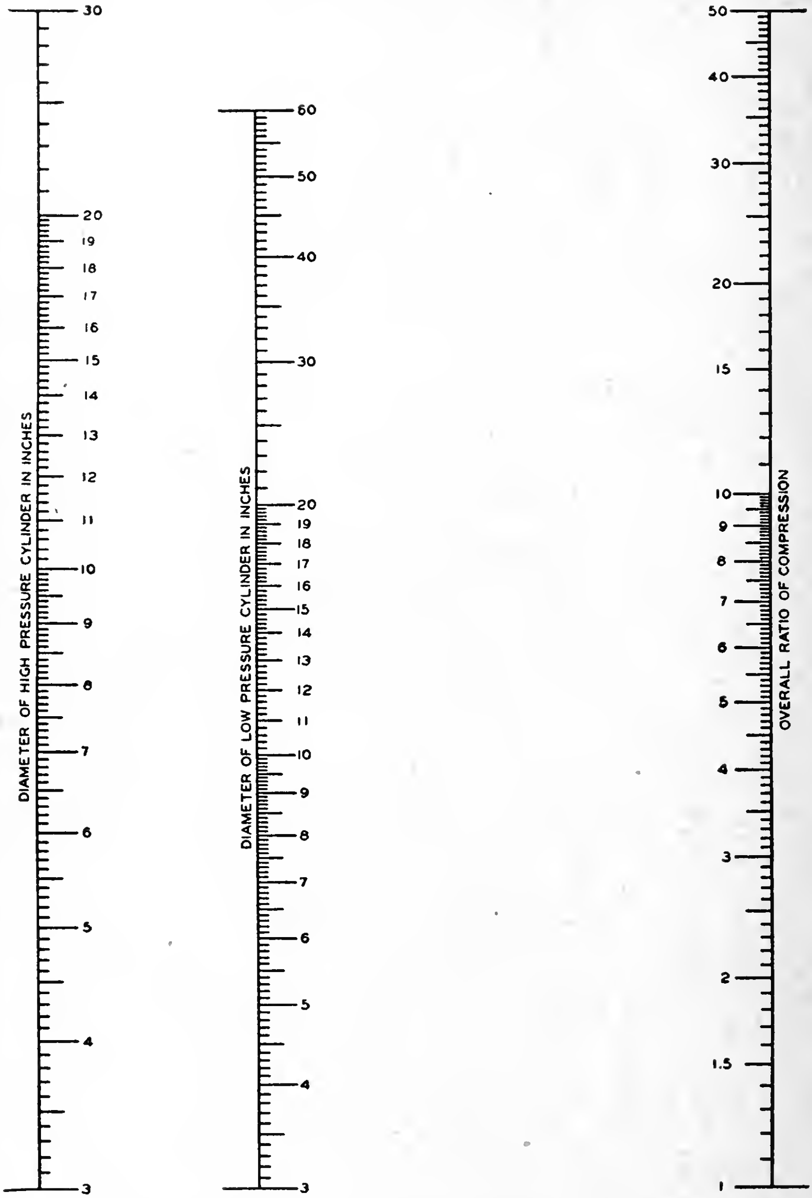


CHART 11. Cylinder Sizes for Two-Stage Compression.
Based on equation 32. See problem 22, Chapter XII.

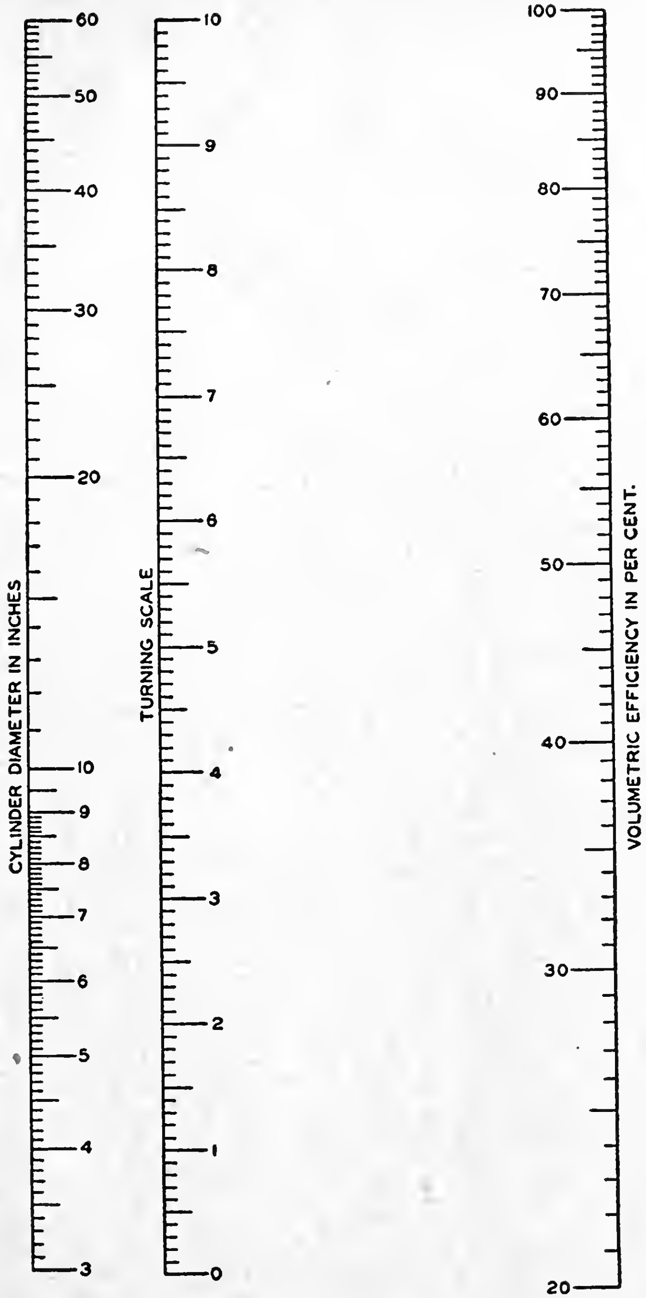


CHART 12. Volumetric Efficiency Correction for Cylinder Size.
Based on equation 50. See problem 23, Chapter XII.

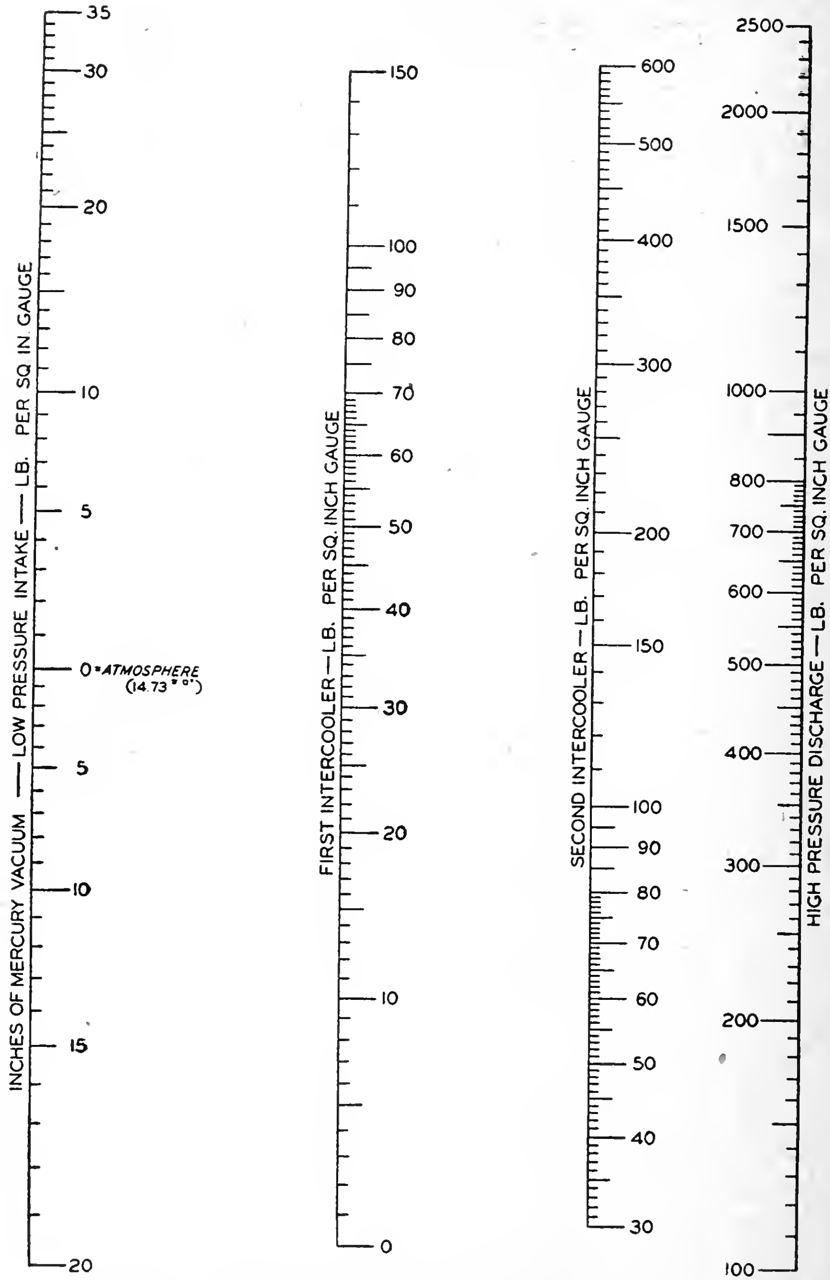


CHART 13. Intercooler Pressures for Three-Stage Compression.
Based on equations 38 and 39. See problem 24, Chapter XII.

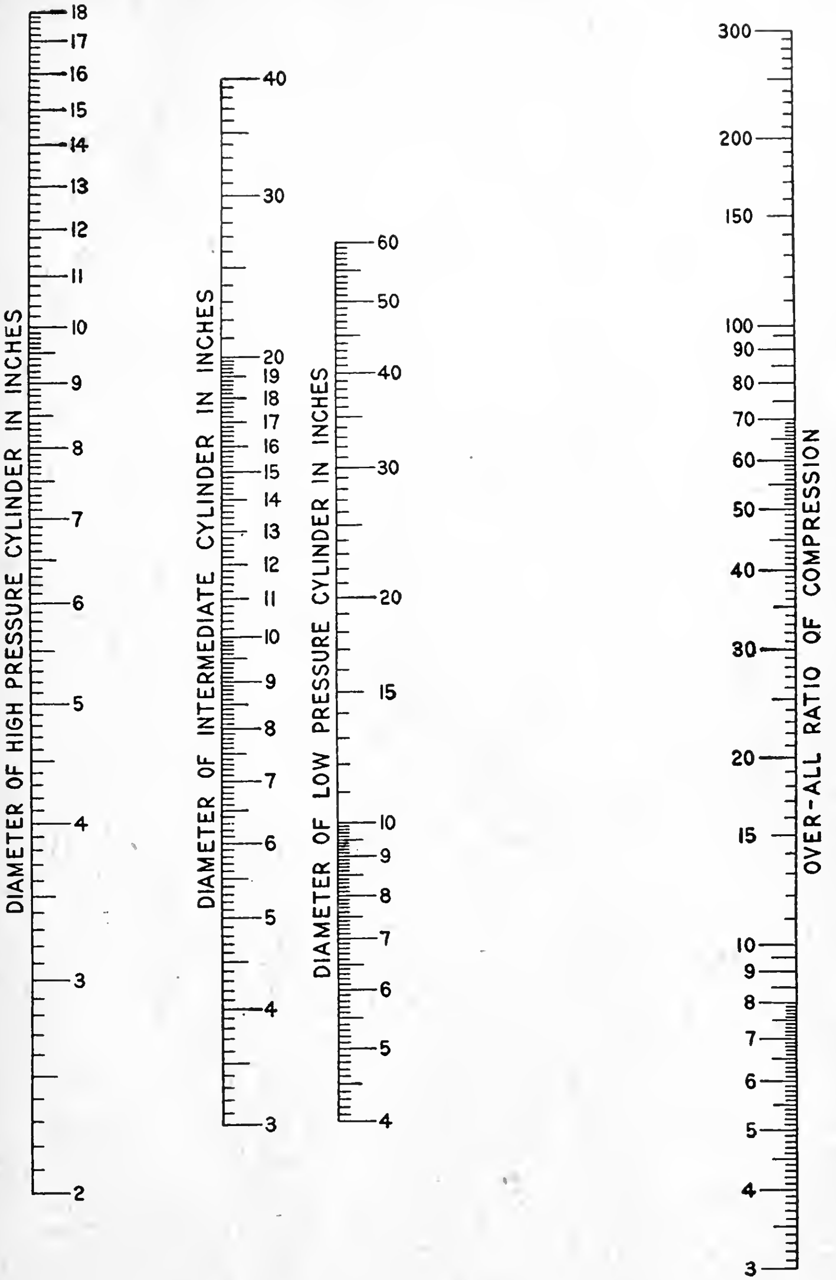


CHART 14. Cylinder Sizes for Three-Stage Compression.
Based on equations 51 and 52. See problem 25, Chapter XII.

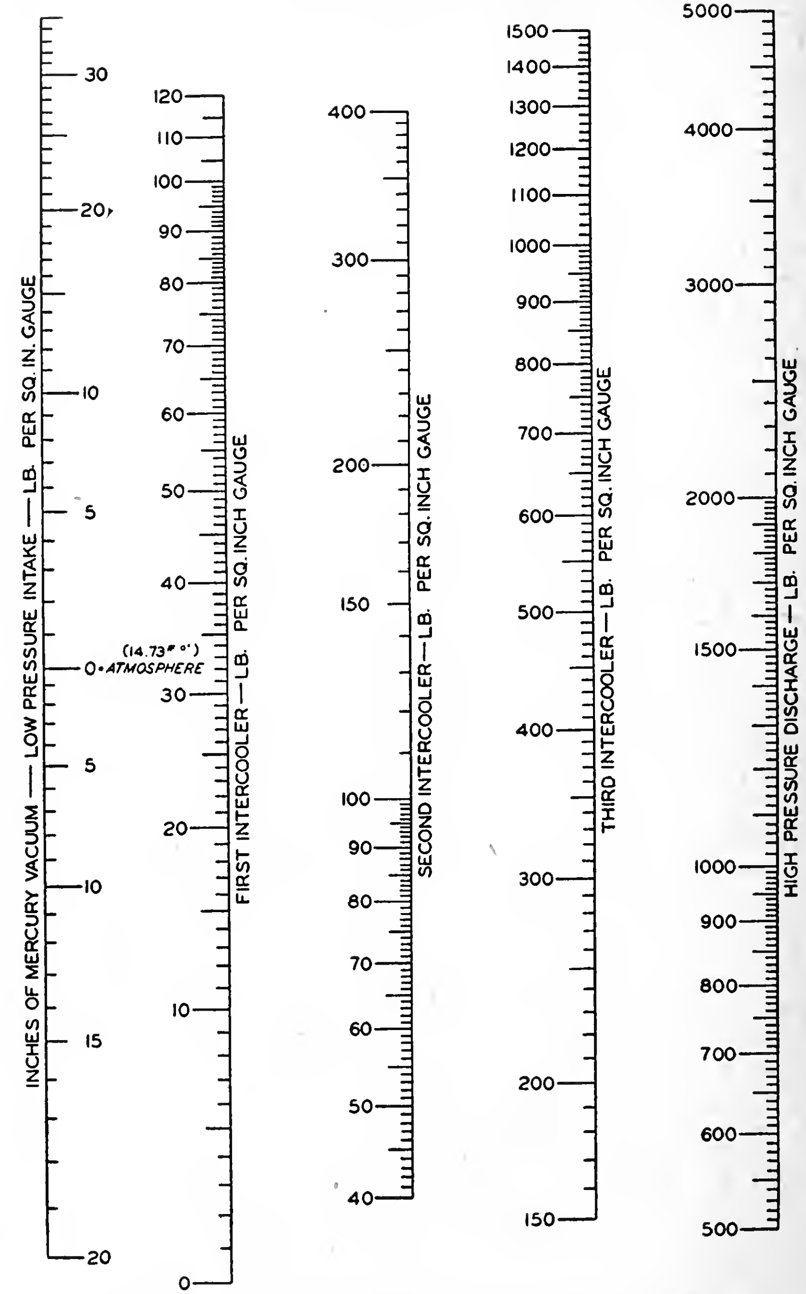


CHART 15. Intercooler Pressures for Four-Stage Compression: *Based on equations 44, 45, and 46. See problem 26, Chapter XII.*

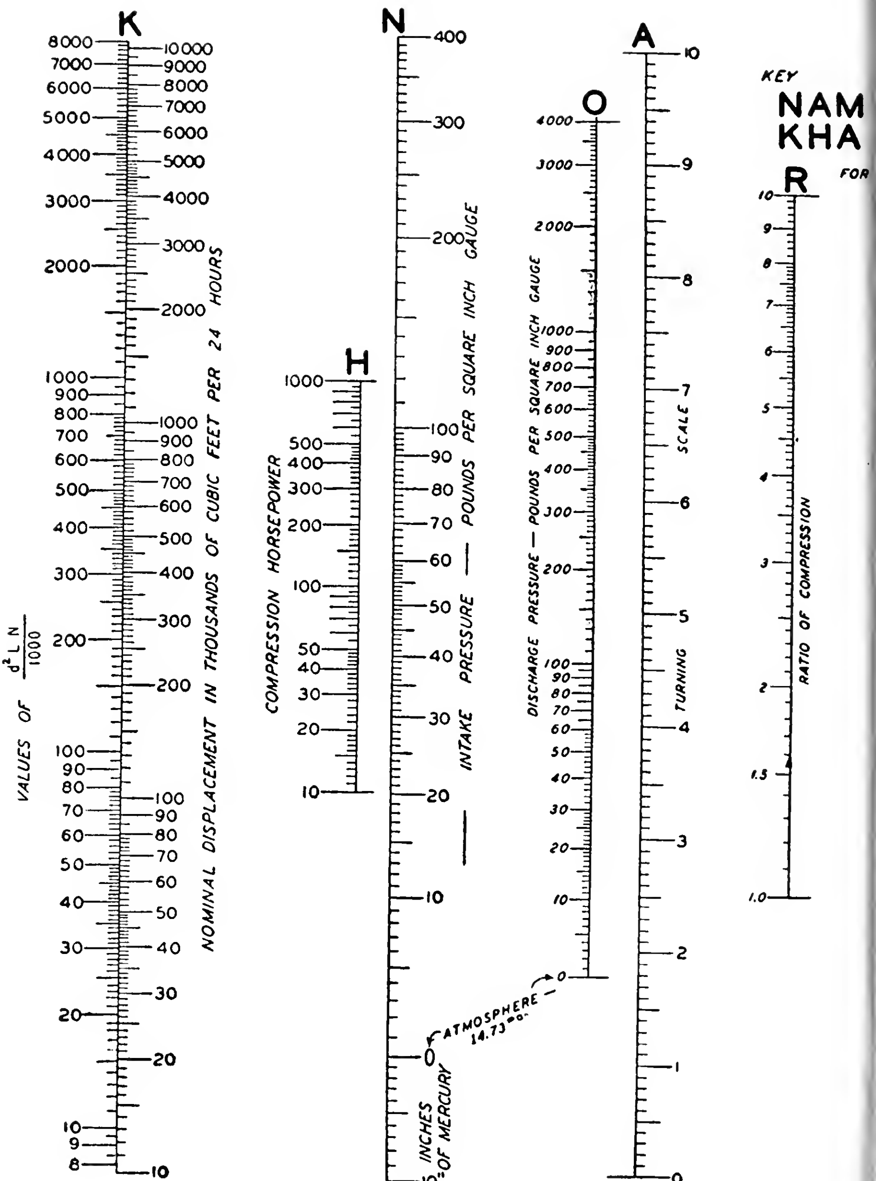
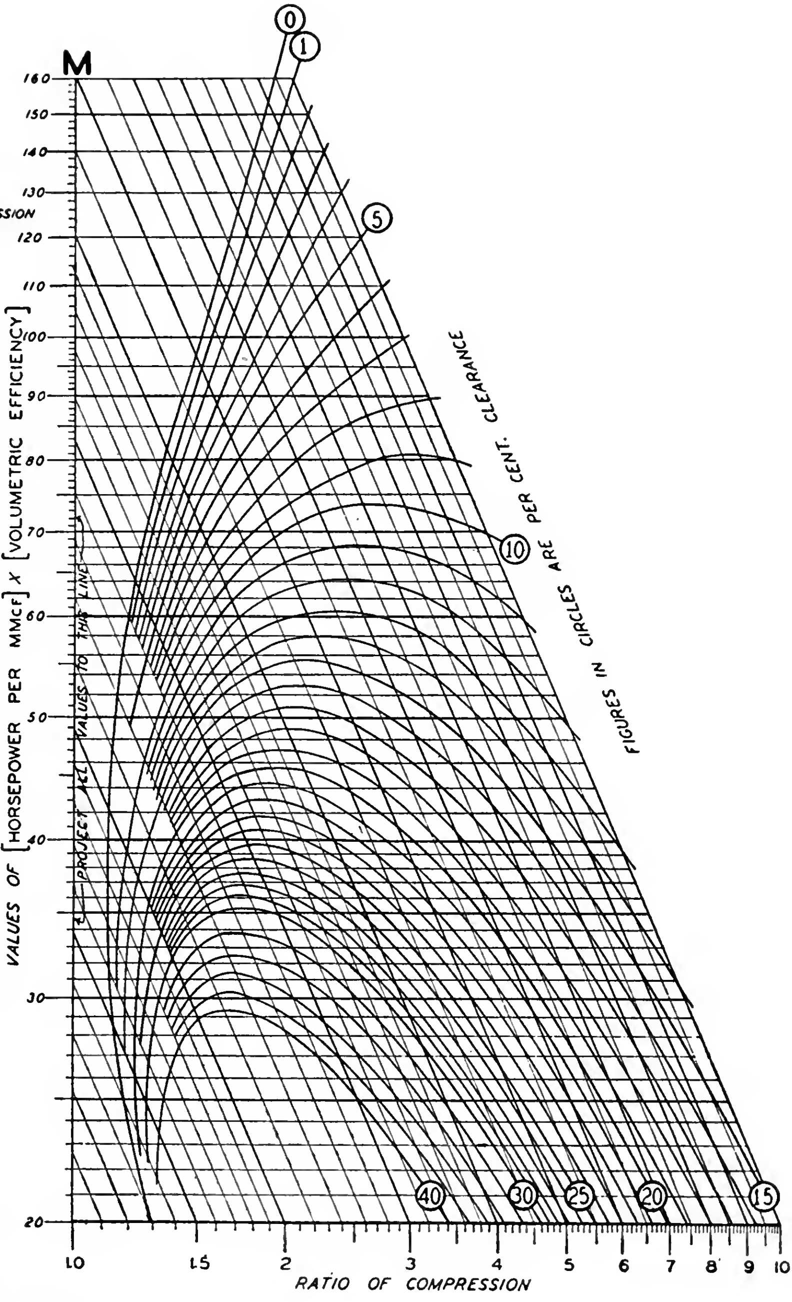


CHART 17. Co
Based on equation 63. See pro

OF COMPRESSION

OR



ssor Unloading.

28, 29, and 30, Chapter XII.

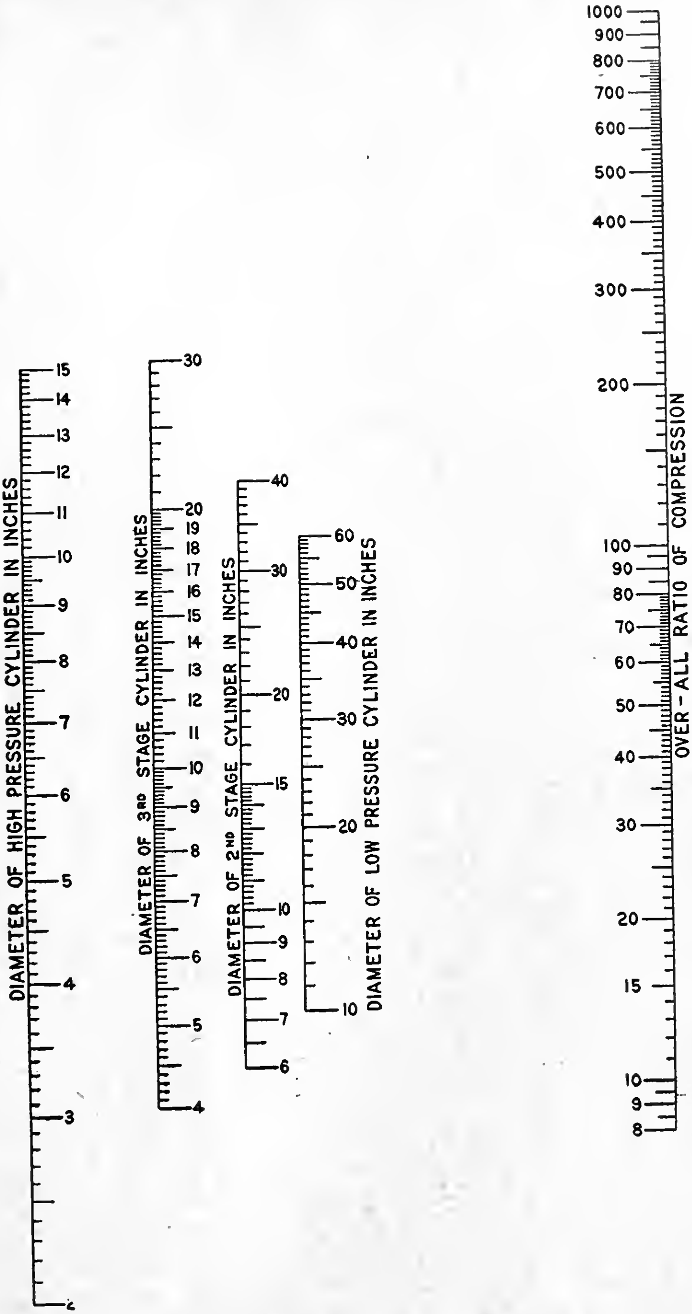


Chart 16. Cylinder Sizes for Four-Stage Compression.
Based on equations 53, 54, and 55. See problem 27, Chapter XII.

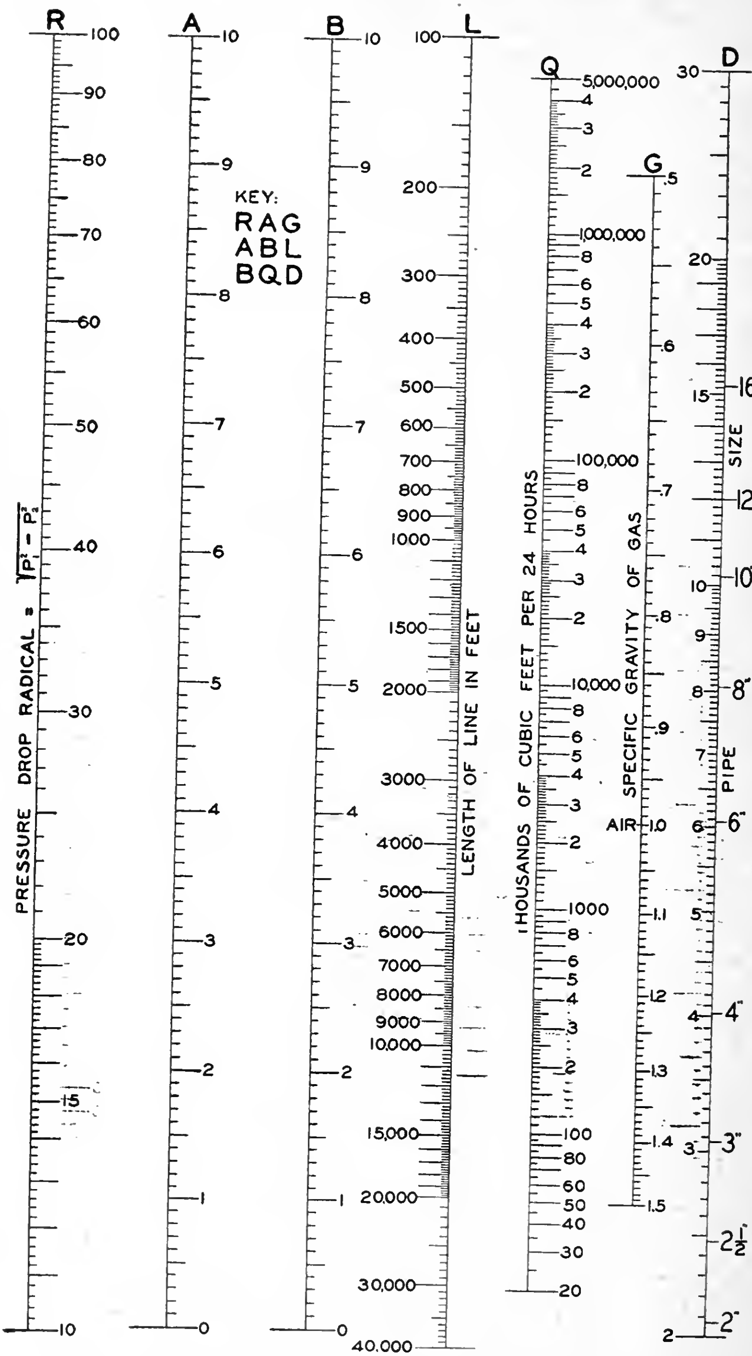


CHART 18. Pipe-Line Pressure Drop.
Based on equation 88. See problems 1, 2, and 3, Chapter XVII.

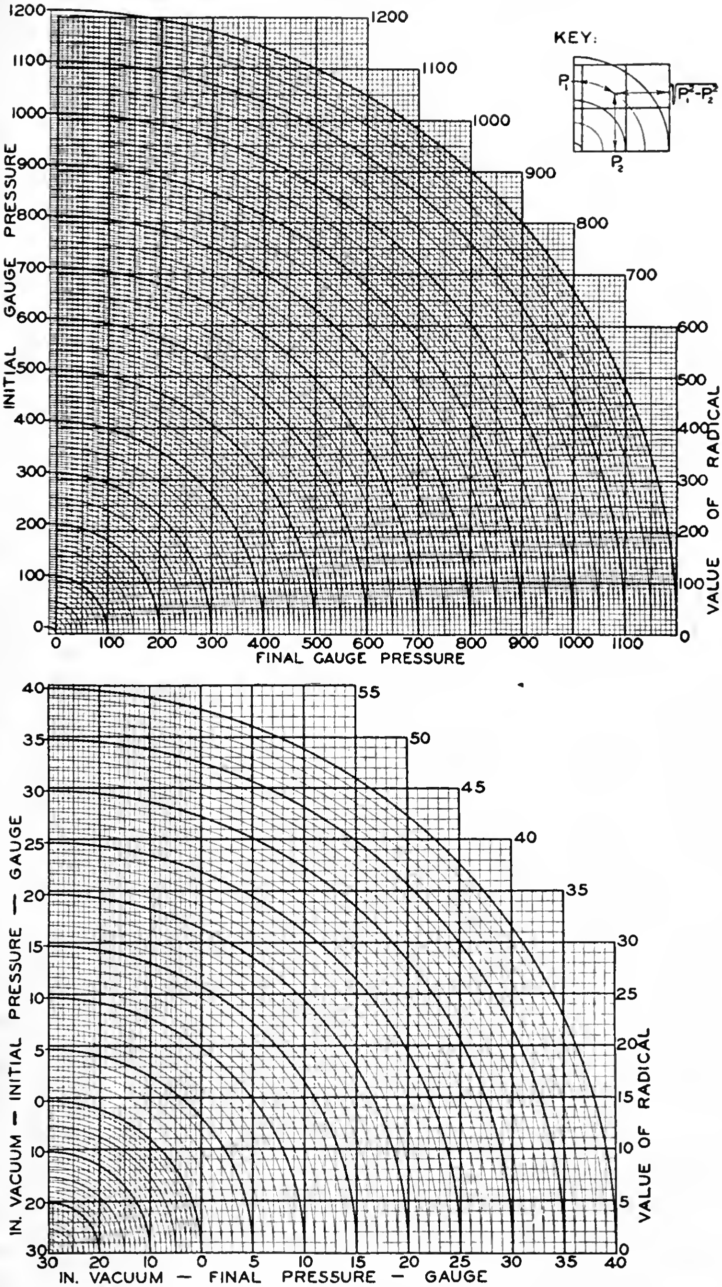


CHART 19. Values of $\sqrt{P_1{}^2 - P_2{}^2}$ for Pipe-Line Pressure-Drop Calculations.
See problems 1, 2, and 3, Chapter XVII.

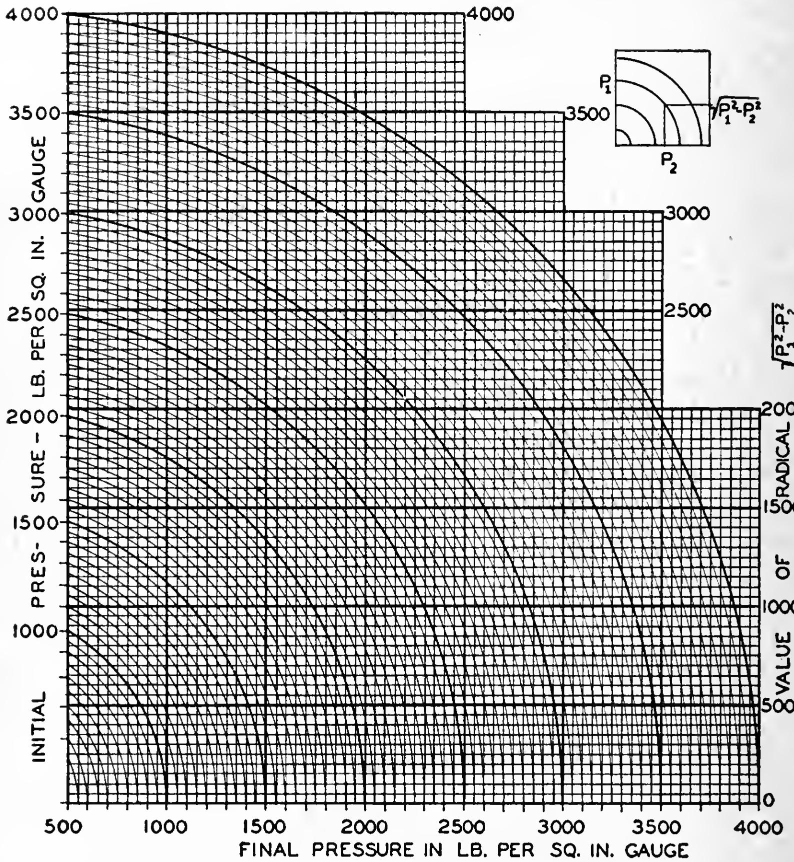


CHART 20. Values of $\sqrt{P_1^2 - P_2^2}$ for Pipe-Line Pressure-Drop Calculations.
See problems 1, 2, and 3, Chapter XVII.

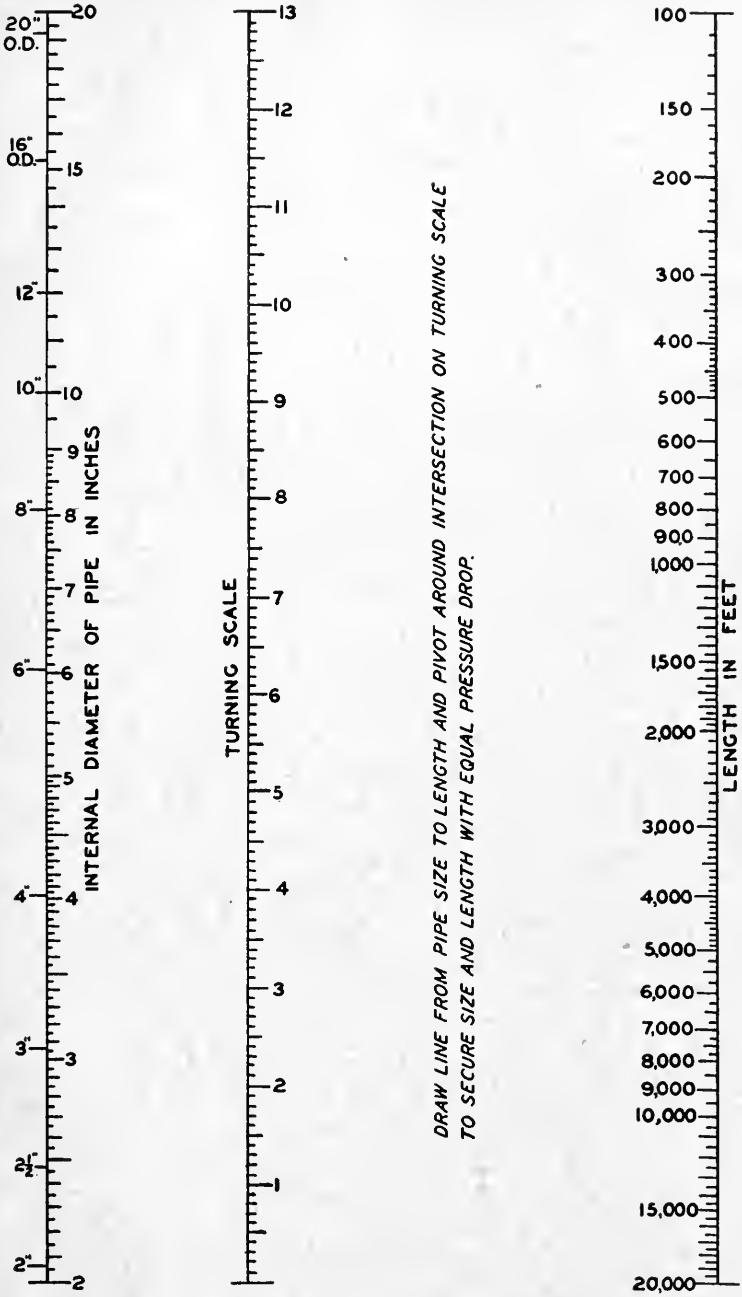


CHART 21. Complex Pipe Systems — Equivalent Length.
Based on equations 93 and 94. See problems 5 and 6, Chapter XVII.

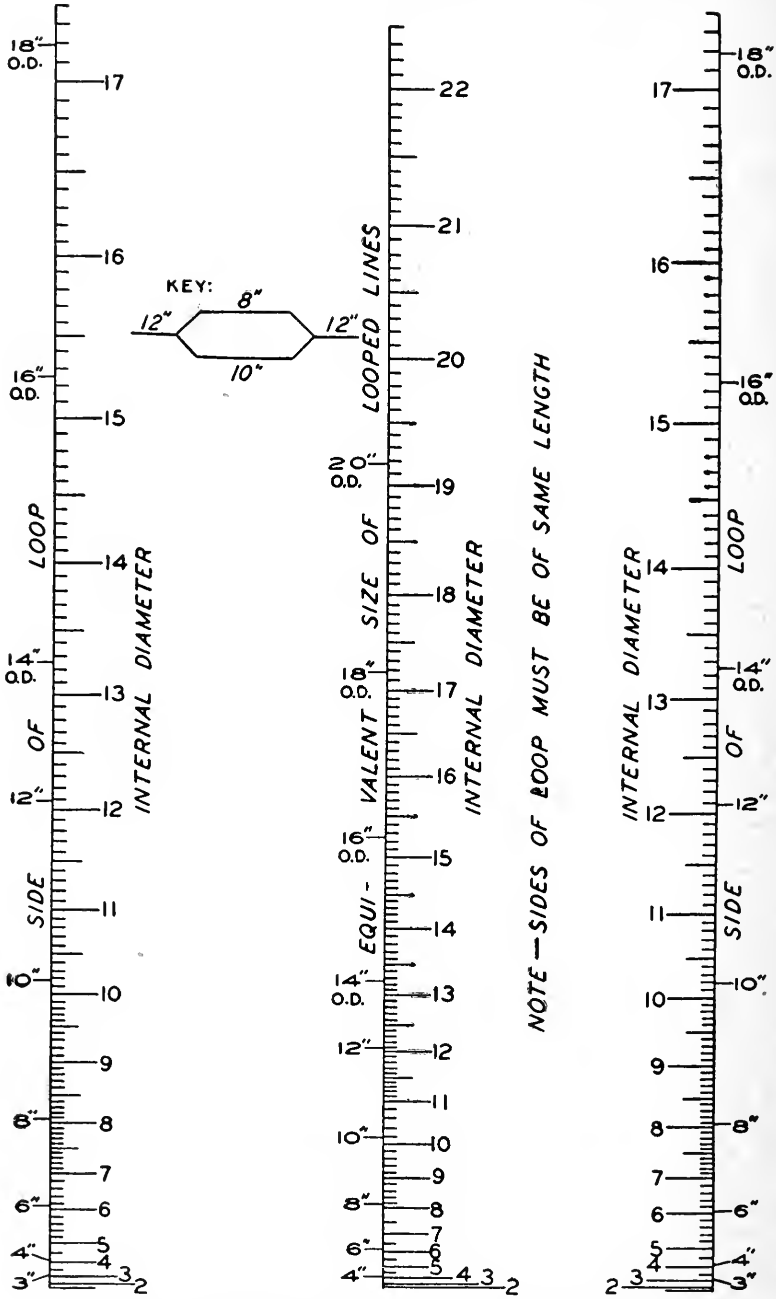


CHART 22. Complex Pipe Systems — Equivalent Size of Loop Lines.
Based on equation 93. See problem 7, Chapter XVII.

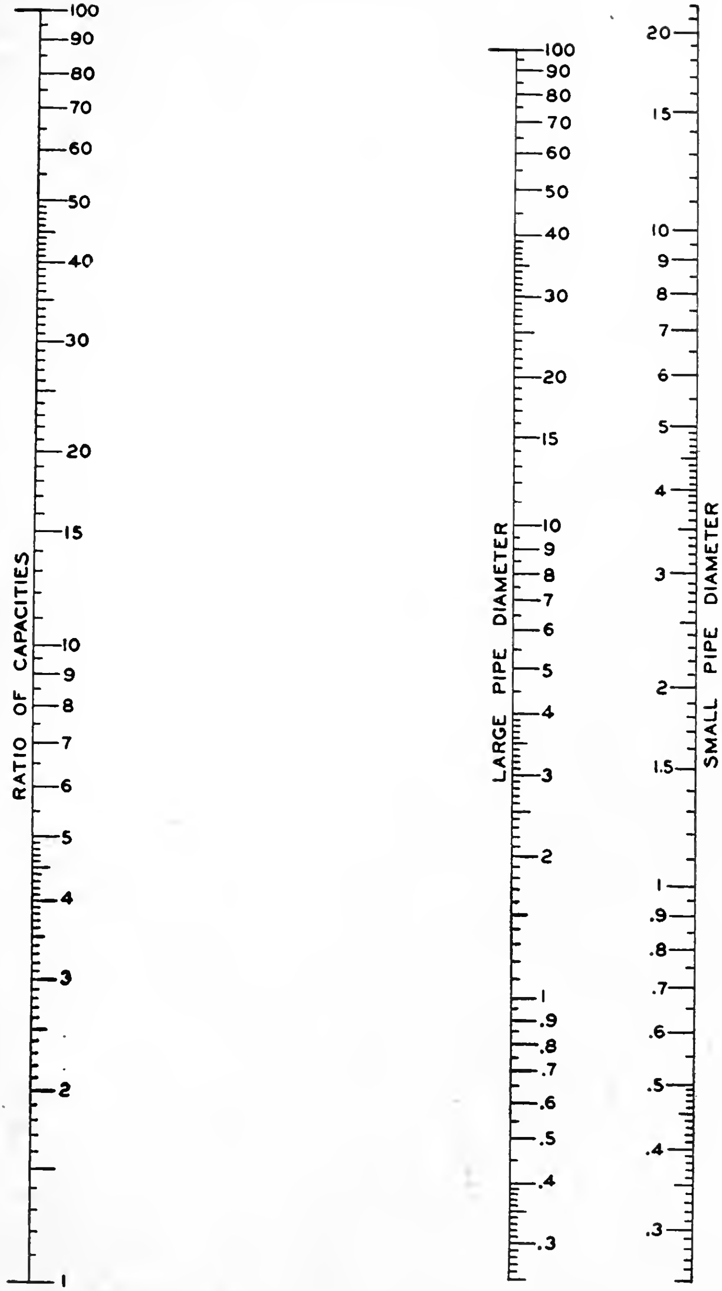


Chart 23. Comparative Capacities of Pipes:
Based on equation 96. See problem 10, Chapter XVII:

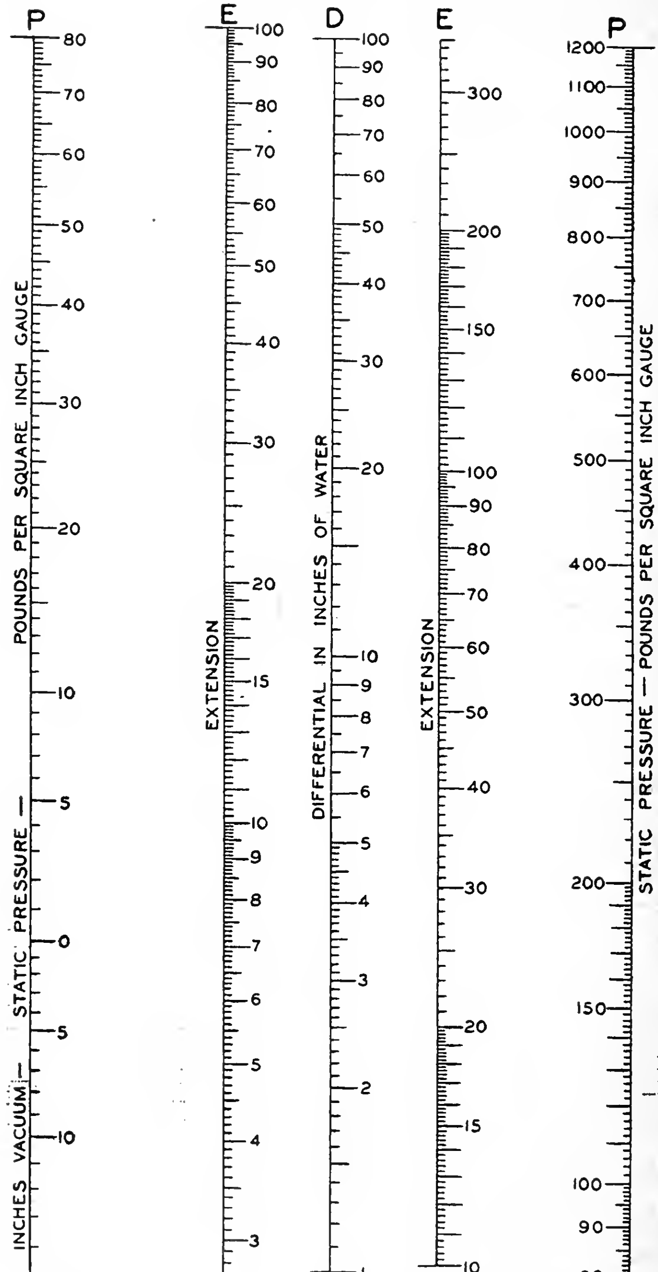
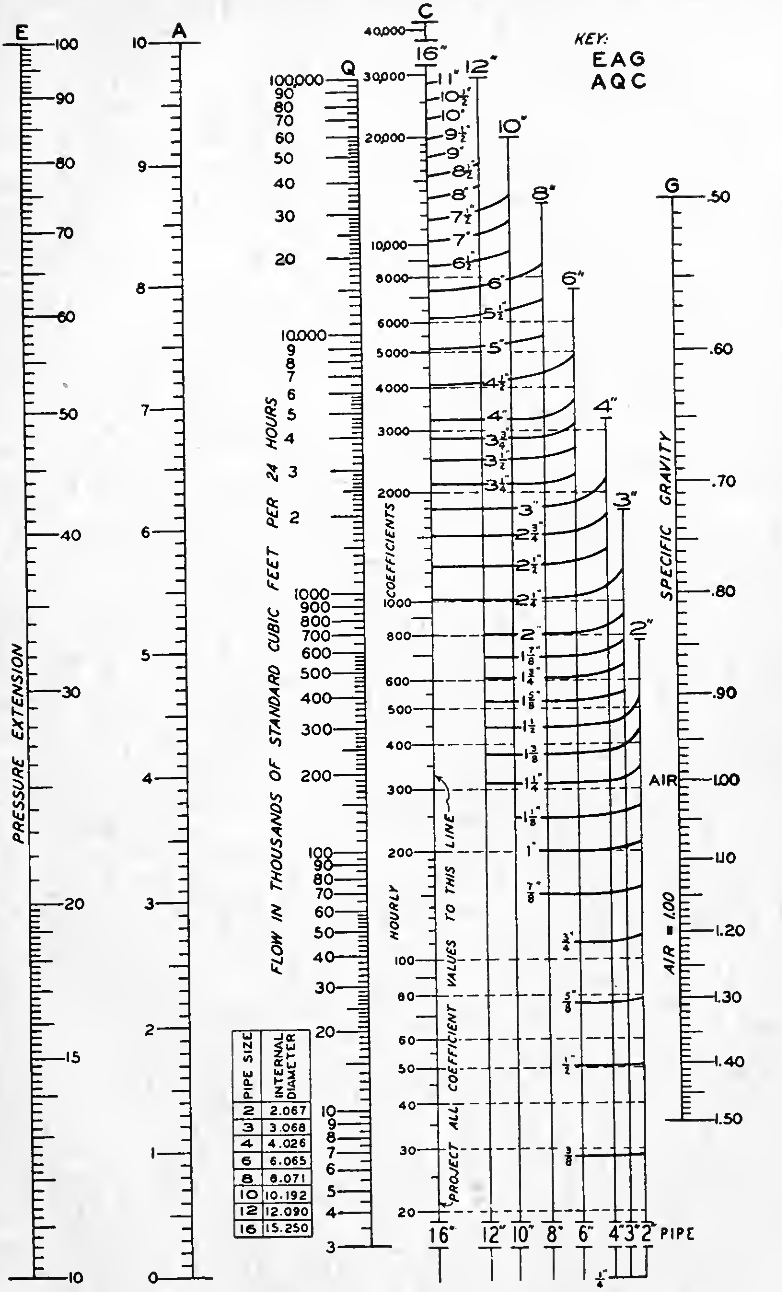


CHART 24. Pressure Extensions for Orifice Meter Calculations.
This chart gives values of $\sqrt{hP}$ in equation 98. See problems 1 and 2, Chapter XVIII.



PIPE SIZE	INTERNAL DIAMETER
2	2.067
3	3.068
4	4.026
6	6.065
8	8.071
10	10.192
12	12.090
16	15.250

CHART 25. Orifice Meters for Gas and Air — Flange Taps.
Based on equation 98. See problem 2, Chapter XVIII.

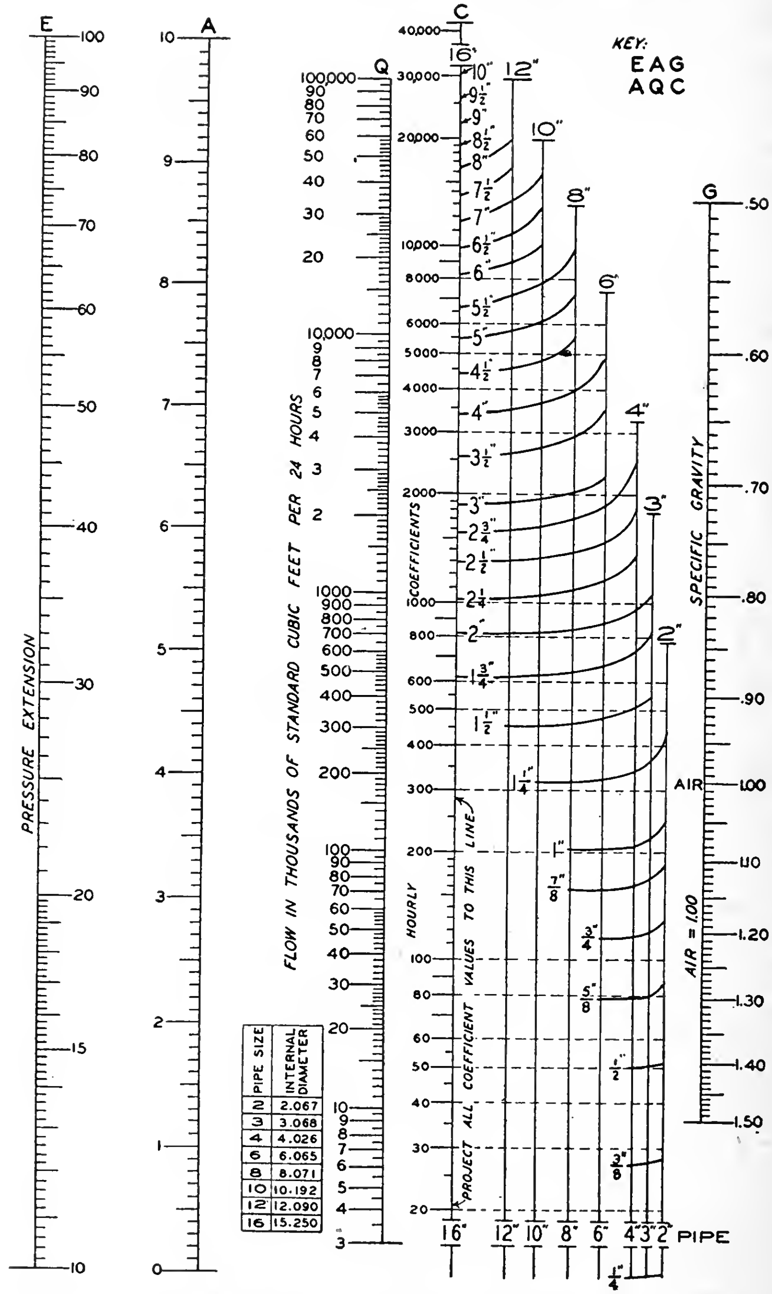
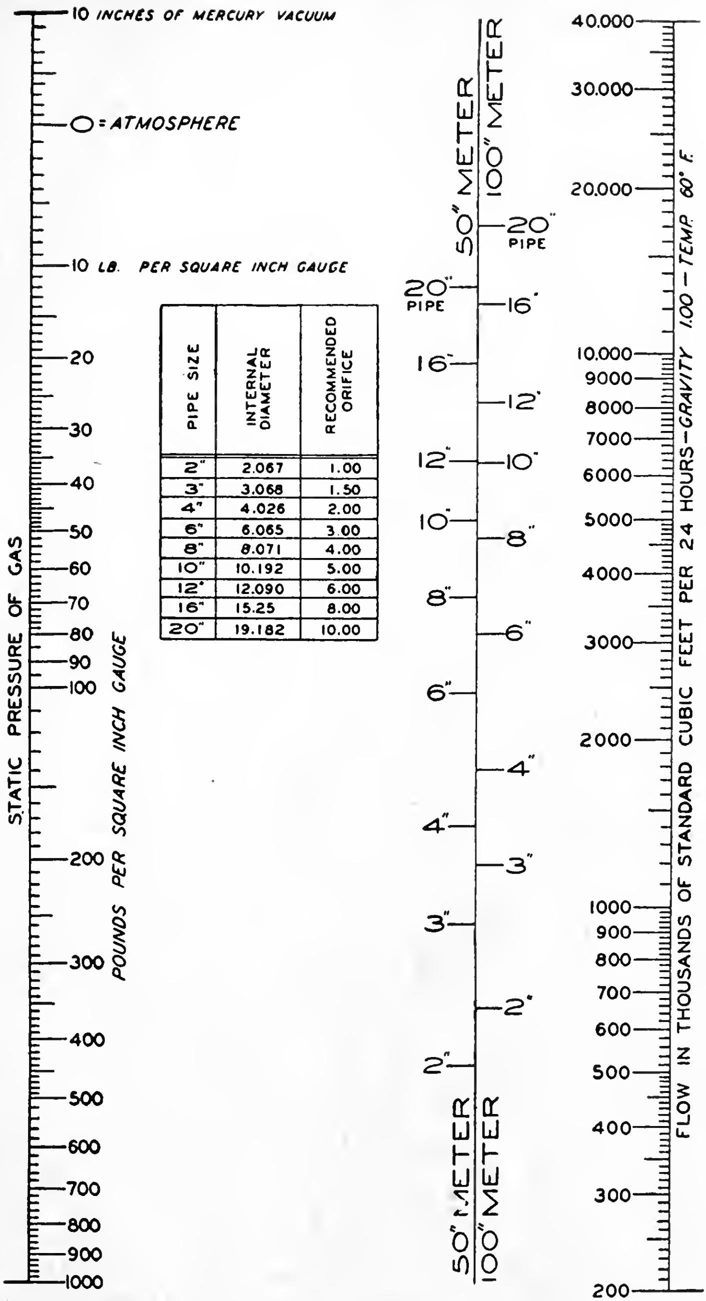


CHART 26. Orifice Meters for Gas and Air — Line ($2\frac{1}{2}$ and 8 diameter) Taps.
Based on equation 98. See problem 1, Chapter XVIII.



PIPE SIZE	INTERNAL DIAMETER	RECOMMENDED ORIFICE
2"	2.067	1.00
3"	3.068	1.50
4"	4.026	2.00
6"	6.065	3.00
8"	8.071	4.00
10"	10.192	5.00
12"	12.090	6.00
16"	15.25	8.00
20"	19.182	10.00

CHART 27. Recommended Sizes for Orifice Meters.
Based on equation 103. See problems 3 and 4, Chapter XVIII.

TABLE III. *Part I.*

14.6-Lb. Pressure Multipliers — 14.73-Lb. Base

Inches of Mercury Vacuum

	0	1	2	3	4	5	6	7	8	9
20	0.324	0.291	0.258	0.224	0.191	0.158	0.124	0.091	0.058	0.024
10	0.658	0.624	0.591	0.558	0.524	0.491	0.458	0.424	0.391	0.358
0	0.991	0.958	0.924	0.890	0.858	0.824	0.791	0.758	0.724	0.691

Pounds per Square Inch Gauge

	0	1	2	3	4	5	6	7	8	9
0	0.99	1.06	1.13	1.19	1.26	1.33	1.40	1.47	1.53	1.60
10	1.67	1.74	1.80	1.87	1.94	2.01	2.08	2.14	2.21	2.28
20	2.35	2.42	2.48	2.55	2.62	2.69	2.76	2.82	2.89	2.96
30	3.03	3.09	3.16	3.23	3.30	3.37	3.43	3.50	3.57	3.64
40	3.70	3.77	3.84	3.91	3.98	4.04	4.11	4.18	4.25	4.32
50	4.38	4.45	4.52	4.59	4.66	4.72	4.79	4.86	4.93	5.00
60	5.06	5.13	5.20	5.27	5.33	5.40	5.47	5.54	5.61	5.67
70	5.74	5.81	5.88	5.95	6.01	6.08	6.15	6.22	6.28	6.35
80	6.42	6.49	6.56	6.62	6.69	6.76	6.83	6.90	6.96	7.03
90	7.10	7.17	7.23	7.30	7.37	7.44	7.51	7.57	7.64	7.71
100	7.78	7.85	7.91	7.98	8.05	8.12	8.19	8.25	8.32	8.39
110	8.46	8.52	8.59	8.66	8.73	8.80	8.86	8.93	9.00	9.07
120	9.14	9.20	9.27	9.34	9.41	9.47	9.54	9.61	9.68	9.75
130	9.81	9.88	9.95	10.02	10.09	10.15	10.22	10.29	10.36	10.43
140	10.49	10.56	10.63	10.70	10.76	10.83	10.90	10.97	11.04	11.10
150	11.17	11.24	11.31	11.38	11.44	11.51	11.58	11.65	11.71	11.78
160	11.85	11.92	11.99	12.05	12.12	12.19	12.26	12.33	12.39	12.46
170	12.53	12.60	12.66	12.73	12.80	12.87	12.94	13.00	13.07	13.14
180	13.21	13.28	13.34	13.41	13.48	13.55	13.62	13.68	13.75	13.82
190	13.89	13.95	14.02	14.09	14.16	14.23	14.29	14.36	14.43	14.50
200	14.57	14.63	14.70	14.77	14.84	14.90	14.97	15.04	15.11	15.18
210	15.24	15.31	15.38	15.45	15.52	15.58	15.65	15.72	15.79	15.85
220	15.92	15.99	16.06	16.13	16.19	16.26	16.33	16.40	16.47	16.53
230	16.60	16.67	16.74	16.81	16.87	16.94	17.01	17.08	17.14	17.21
240	17.28	17.35	17.42	17.48	17.55	17.62	17.69	17.76	17.82	17.89

TABLE III. *Part I* (*continued*)

	Pounds per Square Inch Gauge									
	0	1	2	3	4	5	6	7	8	9
250	17.96	18.03	18.09	18.16	18.23	18.30	18.37	18.43	18.50	18.57
260	18.64	18.71	18.77	18.84	18.91	18.98	19.06	19.11	19.18	19.25
270	19.32	19.38	19.45	19.52	19.59	19.66	19.72	19.79	19.86	19.93
280	20.00	20.06	20.13	20.20	20.77	20.33	20.40	20.47	20.54	20.61
290	20.67	20.74	20.81	20.88	20.95	21.01	21.08	21.15	21.22	21.28
300	21.35	21.42	21.49	21.56	21.62	21.69	21.76	21.83	21.90	21.96
310	22.03	22.10	22.17	22.24	22.30	22.37	22.44	22.51	22.57	22.64
320	22.71	22.78	22.85	22.91	22.98	23.05	23.12	23.19	23.25	23.32
330	23.39	23.46	23.52	23.59	23.66	23.73	23.80	23.86	23.93	24.00
340	24.07	24.14	24.20	24.27	24.34	24.41	24.48	24.54	24.61	24.68
350	24.75	24.81	24.88	24.95	25.02	25.09	25.15	25.22	25.29	25.36
360	25.43	25.49	25.56	25.63	25.70	25.76	25.83	25.90	25.97	26.04
370	26.10	26.17	26.24	26.31	26.38	26.44	26.51	26.58	26.65	26.71
380	26.78	26.85	26.92	26.99	27.05	27.12	27.19	27.26	27.33	27.39
390	27.46	27.53	27.60	27.67	27.73	27.80	27.87	27.94	28.00	28.07
400	28.14	28.21	28.28	28.34	28.41	28.48	28.55	28.62	28.68	28.75
410	28.82	28.89	28.95	29.02	29.09	29.16	29.23	29.29	29.36	29.43
420	29.49	29.57	29.62	29.70	29.77	29.84	29.91	29.97	30.04	30.11
430	30.18	30.24	30.31	30.38	30.45	30.52	30.58	30.65	30.72	30.79
440	30.86	30.92	30.99	31.06	31.13	31.19	31.26	31.33	31.40	31.47
450	31.53	31.60	31.67	31.74	31.81	31.87	31.94	32.01	32.08	32.14
460	32.21	32.28	32.35	32.42	32.48	32.55	32.62	32.69	32.76	32.82
470	32.89	32.96	33.03	33.10	33.16	33.23	33.30	33.37	33.43	33.50
480	33.57	33.64	33.71	33.77	33.84	33.91	33.98	34.05	34.11	34.18
490	34.25	34.32	34.38	34.45	34.52	34.59	34.66	34.72	34.79	34.86
500	34.93	35.00	35.06	35.13	35.20	25.27	35.34	35.40	35.47	35.54

$$\text{Factor} = \frac{P + 14.6}{14.73}$$

where P = gauge pressure of gas in pounds per square inch.

See Chapter XIII for deviation factors.

TABLE III. *Part II*

14.6-Lb. Pressure Multipliers — 14.73-Lb. Base

	Pounds per Square Inch Gauge									
	0	10	20	30	40	50	60	70	80	90
0	0.99	1.67	2.35	3.03	3.71	4.38	5.06	5.74	6.42	7.10
100	7.78	8.46	9.14	9.81	10.49	11.17	11.85	12.53	13.21	13.89
200	14.57	15.24	15.92	16.60	17.28	17.96	18.64	19.32	20.00	20.68
300	21.35	22.03	22.71	23.39	24.07	24.75	25.46	26.10	26.78	27.46
400	28.14	28.82	29.50	30.18	30.86	31.54	32.21	32.89	33.57	34.25
500	34.93	35.61	36.29	36.97	37.65	38.33	39.01	39.69	40.37	41.04
600	41.72	42.40	43.08	43.76	44.44	45.12	45.80	46.48	47.16	47.83
700	48.51	49.19	49.87	50.55	51.23	51.91	52.59	53.26	53.94	54.62
800	55.30	55.98	56.66	57.34	58.02	58.70	59.38	60.05	60.73	61.41
900	62.09	62.77	63.45	64.13	64.81	65.48	66.16	66.84	67.52	68.20
1000	68.88	69.56	70.24	70.92	71.60	72.27	72.95	73.63	74.31	74.99
1100	75.67	76.35	77.02	77.70	78.38	79.06	79.74	80.42	81.10	81.78
1200	82.46	83.14	83.82	84.49	85.17	85.85	86.53	87.21	87.89	88.57
1300	89.25	89.92	90.60	91.28	91.96	92.64	93.32	94.00	94.68	95.35
1400	96.03	96.71	97.39	98.07	98.75	99.43	100.1	100.8	101.5	102.1
1500	102.8	103.5	104.2	104.9	105.5	106.2	106.9	107.6	108.2	108.9
1600	109.6	110.3	111.0	111.6	112.3	113.0	113.7	114.4	115.0	115.7
1700	116.4	117.1	117.8	118.4	119.1	119.8	120.5	121.1	121.8	122.5
1800	123.2	123.9	124.5	125.2	125.9	126.6	127.3	127.9	128.6	129.3
1900	130.0	130.7	131.3	132.0	132.7	133.4	134.0	134.7	135.4	136.1

TABLE III. *Part II (continued)*

	Pounds per Square Inch Gauge									
	0	10	20	30	40	50	60	70	80	90
2000	136.8	137.4	138.1	138.8	139.5	140.2	140.8	141.5	142.2	142.9
2100	143.5	144.2	144.9	145.6	146.3	146.9	147.6	148.3	149.0	149.7
2200	150.3	151.0	151.7	152.4	153.1	153.7	154.4	155.1	155.8	156.4
2300	157.1	157.8	158.5	159.2	159.8	160.5	161.2	161.9	162.6	163.2
2400	163.9	164.6	165.3	166.0	166.6	167.3	168.0	168.7	169.4	170.0
2500	170.7	171.4	172.1	172.7	173.4	174.1	174.8	175.5	176.1	176.8
2600	177.5	178.2	178.9	179.5	180.2	180.9	181.6	182.2	182.9	183.6
2700	184.3	185.0	185.6	186.3	187.0	187.7	188.4	189.0	189.7	190.4
2800	191.1	191.8	192.4	193.1	193.8	194.5	195.1	195.8	196.5	197.2
2900	197.9	198.5	199.2	199.9	200.6	201.3	201.9	202.6	203.3	204.0
3000	204.7	205.3	206.0	206.7	207.4	208.0	208.7	209.4	210.1	210.8
3100	211.4	212.1	212.8	213.5	214.2	214.8	215.5	216.2	216.9	217.5
3200	218.2	218.9	219.6	220.3	220.9	221.6	222.3	223.0	223.7	224.3
3300	225.0	225.7	226.4	227.1	227.7	228.4	229.1	229.8	230.4	231.1
3400	231.8	232.5	233.2	233.8	234.5	235.2	235.9	236.6	237.2	237.9
3500	238.6	239.3	240.0	240.6	241.3	242.0	242.7	243.3	244.0	244.7
3600	245.4	246.1	246.7	247.4	248.1	248.8	249.5	250.1	250.8	251.5
3700	252.2	252.9	253.5	254.2	254.9	255.6	256.2	256.9	257.6	258.3
3800	259.0	259.6	260.3	261.0	261.7	262.4	263.0	263.7	264.4	265.1
3900	265.8	266.4	267.1	267.8	268.5	269.1	269.8	270.5	271.2	271.9
4000	272.5	273.2	273.9	274.6	275.3	275.9	276.6	277.3	278.0	278.7

$$\text{Factor} = \frac{P + 14.6}{14.73}$$

where P = gauge pressure of gas in pounds per square inch.

See Chapter XIII for deviation factors.

TABLE IV

THEORETICAL COMPRESSION HORSEPOWER FACTORS FOR VOLUMES IN STANDARD CUBIC FEET PER MINUTE

Values of R	Isothermal	Polytropics				
	$n = 1.0$	$n = 1.1$	$n = 1.2$	$n = 1.3$	$n = 1.4$	$n = 1.5$
1.4	0.0216	0.0219	0.0222	0.0225	0.0227	0.0229
1.6	0.0301	0.0308	0.0314	0.0318	0.0323	0.0327
1.8	0.0377	0.0388	0.0398	0.0405	0.0412	0.0417
2.0	0.0445	0.0459	0.0472	0.0482	0.0492	0.0501
2.2	0.0505	0.0525	0.0542	0.0556	0.0568	0.0581
2.4	0.0563	0.0586	0.0606	0.0623	0.0640	0.0654
2.6	0.0613	0.0642	0.0665	0.0687	0.0705	0.0723
2.8	0.0662	0.0676	0.0721	0.0748	0.0769	0.0790
3.0	0.0705	0.0743	0.0775	0.0804	0.0829	0.0852
3.2	0.0748	0.0790	0.0825	0.0858	0.0887	0.0913
3.4	0.0786	0.0833	0.0873	0.0908	0.0942	0.0971
3.6	0.0823	0.0873	0.0917	0.0957	0.0994	0.1027
3.8	0.0858	0.0912	0.0960	0.1005	0.1044	0.1080
4.0	0.0890	0.0949	0.1002	0.1049	0.1093	0.1133
4.5	0.0966	0.1035	0.1099	0.1156	0.1208	0.1256
5.0	0.1034	0.1115	0.1187	0.1253	0.1309	0.1369
6.0	0.1152	0.1251	0.1342	0.1426	0.1505	0.1576
8.0	0.1336	0.1472	0.1597	0.1715	0.1827	0.1928
10.0	0.1481	0.1647	0.1804	0.1952	0.2092	0.2226
15.0	0.1740	0.1974	0.2200	0.2418	0.2627	0.2827
20.0	0.1925	0.2213	0.2497	0.2775	0.3046	0.3306
25.0	0.2069	0.2398	0.2737	0.3069	0.3394	0.3709

This table is for the solution of equations 11 and 16.

For isothermal compression ($n = 1.0$), factor $= 0.1479 \log R$.

For polytropics, factor $= \dfrac{0.0643\, n}{n - 1} [R^{(n-1)/n} - 1]$.

To obtain theoretical compression horsepower, multiply factor by gas to be compressed in standard cubic feet per minute.

TABLE V

Theoretical Compression Horsepower Factors for Volumes in Thousands of Cubic Feet per 24 Hours

Values of R	Isothermal	Polytropics				
	$n = 1.0$	$n = 1.1$	$n = 1.2$	$n = 1.3$	$n = 1.4$	$n = 1.5$
1.4	0.0150	0.0152	0.0154	0.0156	0.0157	0.0159
1.6	0.0209	0.0214	0.0218	0.0221	0.0224	0.0227
1.8	0.0262	0.0269	0.0276	0.0281	0.0286	0.0290
2.0	0.0309	0.0319	0.0328	0.0335	0.0342	0.0348
2.2	0.0351	0.0365	0.0376	0.0386	0.0395	0.0403
2.4	0.0391	0.0407	0.0421	0.0433	0.0444	0.0454
2.6	0.0426	0.0446	0.0462	0.0477	0.0490	0.0502
2.8	0.0459	0.0469	0.0501	0.0519	0.0534	0.0548
3.0	0.0490	0.0516	0.0538	0.0558	0.0576	0.0592
3.2	0.0519	0.0548	0.0573	0.0596	0.0616	0.0634
3.4	0.0546	0.0578	0.0606	0.0631	0.0654	0.0674
3.6	0.0572	0.0606	0.0637	0.0665	0.0690	0.0713
3.8	0.0596	0.0633	0.0667	0.0698	0.0725	0.0750
4.0	0.0618	0.0659	0.0696	0.0729	0.0759	0.0787
4.5	0.0671	0.0719	0.0763	0.0803	0.0839	0.0872
5.0	0.0718	0.0774	0.0824	0.0870	0.0909	0.0951
6.0	0.0800	0.0869	0.0932	0.0990	0.1044	0.1094
8.0	0.0928	0.1022	0.1109	0.1191	0.1268	0.1339
10.0	0.1028	0.1143	0.1253	0.1356	0.1454	0.1546
15.0	0.1208	0.1371	0.1528	0.1679	0.1824	0.1963
20.0	0.1337	0.1537	0.1734	0.1927	0.2115	0.2296
25.0	0.1437	0.1665	0.1901	0.2131	0.2357	0.2576

This table is for the solution of equations 12 and 17.

For isothermal compression ($n = 1.0$), factor $= 0.10275 \log R$.

For polytropics, factor $= \dfrac{0.0446\,n}{n-1}\,[R^{(n-1)/n} - 1]$.

To obtain theoretical compression horsepower, multiply factor by gas to be compressed in thousands of cubic feet per 24 hours (M.c.f.).

TABLE VI

Nominal Compressor Displacement Factors—20-In. Stroke and 200 R.P.M.

Cylinder Diameter Inches	A = Effective Area in Sq. In. $\frac{2\pi d^2}{4}$	$\frac{LAN}{33,000}$ L = feet A = sq.in. N = r.p.m.	Cubic Feet per Minute 200 r.p.m.	M. C. F. per 24 Hours 200 r.p.m.	Cylinder Diameter Inches	A = Effective Area in Sq. In. $\frac{2\pi d^2}{4}$	$\frac{LAN}{33,000}$ L = feet A = sq.in. N = r.p.m.	Cubic Feet per Minute 200 r.p.m.	M. C. F. per 24 Hours 200 r.p.m.
3	14.1	0.143	32.6	47.2	15½	377.4	3.812	873.6	1258.0
3¼	16.6	0.168	38.4	55.3	16	402.1	4.062	930.9	1340.4
3½	19.2	0.195	44.6	64.2	16½	427.6	4.310	987.6	1422.2
3¾	22.1	0.224	51.2	73.7	17	454.0	4.586	1050.9	1513.2
4	25.1	0.254	58.2	83.8	17½	481.1	4.860	1113.6	1603.6
4¼	28.4	0.287	65.7	94.6	18	508.9	5.141	1178.1	1696.5
4½	31.8	0.322	73.7	106.1	18½	537.6	5.431	1244.5	1792.0
4¾	35.4	0.358	82.1	118.2	19	567.1	5.728	1312.6	1890.2
5	39.3	0.397	90.9	130.9	19½	597.3	6.034	1382.7	1991.0
5¼	43.2	0.438	100.2	144.4	20	628.3	6.347	1454.5	2094.4
5½	47.5	0.480	110.0	158.4	20½	660.1	6.668	1528.1	2200.4
5¾	51.9	0.525	120.2	173.1	21	692.7	6.998	1603.5	2308.5
6	56.5	0.571	130.9	188.5	21½	726.1	7.335	1680.8	2420.4
6¼	61.4	0.620	142.1	204.6	22	760.3	7.680	1759.9	2534.2
6½	66.4	0.671	153.7	221.3	22½	795.2	8.033	1840.8	2650.8
6¾	71.6	0.723	165.7	238.6	23	831.0	8.394	1923.5	2769.6
7	77.0	0.778	178.2	256.6	23½	867.5	8.763	2008.1	2891.6
7¼	82.5	0.834	191.1	275.2	24	904.8	9.140	2094.4	3016.0
7½	88.4	0.892	204.5	294.5	24½	942.9	9.524	2182.6	3143.0
7¾	94.3	0.953	218.4	314.5	25	981.7	9.917	2272.6	3272.5
8	100.5	1.016	232.7	335.1	25½	1021.4	10.318	2364.4	3404.8
8¼	106.9	1.080	247.5	356.4	26	1061.9	10.726	2458.0	3539.6
8½	113.5	1.147	262.7	378.3	26½	1103.1	11.143	2553.5	3677.0
9	127.2	1.285	294.5	424.1	27	1145.1	11.567	2650.8	3817.1
9½	141.8	1.432	328.2	472.6	27½	1187.9	12.000	2749.8	3959.8
10	157.1	1.587	363.6	523.6	28	1231.5	12.440	2850.7	4105.0
10½	173.2	1.750	400.9	577.3	28½	1275.9	12.888	2953.4	4252.9
11	190.1	1.920	440.0	633.6	29	1321.0	13.344	3058.0	4403.5
11½	207.7	2.098	480.8	692.4	29½	1367.0	13.808	3164.3	4556.6
12	226.2	2.285	523.6	754.0	30	1413.7	14.280	3272.5	4712.4
12½	245.4	2.480	568.2	818.2	31	1509.6	15.249	3494.5	5032.0
13	264.5	2.672	612.2	881.6	32	1608.5	16.248	3723.4	5361.7
13½	286.3	2.892	662.7	954.3	33	1710.6	17.279	3959.7	5702.0
14	307.9	3.110	712.7	1026.3	34	1815.8	18.342	4203.4	6052.8
14½	330.3	3.336	764.5	1100.9	35	1924.2	19.437	4454.2	6414.0
15	353.4	3.570	818.1	1178.1	36	2035.8	20.564	4712.5	6786.0

PISTON RODS

Rod Diameter in Inches	Area In Square Inches	$\frac{LAN}{33,000}$	Cubic Feet per Minute	M. C. F.
1	.78	0.008	1.8	2.6
1¼	1.23	0.012	2.8	4.2
1½	1.77	0.018	4.1	5.9
1¾	2.40	0.024	5.6	8.0
2	3.14	0.032	7.3	10.5
2¼	3.98	0.040	9.2	13.3
2½	4.91	0.050	11.4	16.4
2¾	5.94	0.060	13.7	19.8
3	7.07	0.071	16.4	23.5

For r.p.m. and stroke factors see Table VII.

TABLE VII

R.P.M. AND STROKE CORRECTION FACTORS FOR COMPRESSOR DISPLACEMENT BASED ON A 20-IN. STROKE AND 200 R.P.M. AS 1.00

R.P.M.	Stroke of Piston											
	6-in.	8-in.	10-in.	12-in.	14-in.	15-in.	16-in.	18-in.	20-in.	24-in.	30-in.	36-in.
60	0.090	0.120	0.150	0.180	0.210	0.225	0.240	0.270	0.300	0.360	0.450	0.540
70	0.105	0.140	0.175	0.210	0.245	0.262	0.280	0.315	0.350	0.420	0.525	0.630
80	0.120	0.160	0.200	0.240	0.280	0.300	0.320	0.360	0.400	0.480	0.600	0.720
90	0.135	0.180	0.225	0.270	0.315	0.337	0.360	0.405	0.450	0.540	0.675	0.810
100	0.150	0.200	0.250	0.300	0.350	0.375	0.400	0.450	0.500	0.600	0.750	0.900
110	0.165	0.220	0.275	0.330	0.385	0.412	0.440	0.495	0.550	0.660	0.825	0.990
120	0.180	0.240	0.300	0.360	0.420	0.450	0.480	0.540	0.600	0.720	0.900	1.080
130	0.195	0.260	0.325	0.390	0.455	0.487	0.520	0.585	0.650	0.780	0.975	1.170
140	0.210	0.280	0.350	0.420	0.490	0.525	0.560	0.630	0.700	0.840	1.050	1.260
150	0.225	0.300	0.375	0.450	0.525	0.562	0.600	0.675	0.750	0.900	1.125	1.350
160	0.240	0.320	0.400	0.480	0.560	0.600	0.640	0.720	0.800	0.960	1.200	1.440
170	0.255	0.340	0.425	0.510	0.595	0.637	0.680	0.765	0.850	1.020	1.275	1.530
180	0.270	0.360	0.450	0.540	0.630	0.675	0.720	0.810	0.900	1.080	1.350	1.620
190	0.285	0.380	0.475	0.570	0.665	0.712	0.760	0.855	0.950	1.140	1.425	1.710
200	0.300	0.400	0.500	0.600	0.700	0.750	0.800	0.900	1.000	1.200	1.500	1.800
210	0.315	0.420	0.525	0.630	0.735	0.787	0.840	0.945	1.050	1.260	1.575	1.890
220	0.330	0.440	0.550	0.660	0.770	0.825	0.880	0.990	1.100	1.320	1.650	1.980
230	0.345	0.460	0.575	0.690	0.805	0.862	0.920	1.035	1.150	1.380	1.725	2.070
240	0.360	0.480	0.600	0.720	0.840	0.900	0.960	1.080	1.200	1.440	1.800	2.160
250	0.375	0.500	0.625	0.750	0.875	0.937	1.000	1.125	1.250	1.500	1.875	2.250
260	0.390	0.520	0.650	0.780	0.910	0.975	1.040	1.170	1.300	1.560	1.950	2.340
270	0.405	0.540	0.675	0.810	0.945	1.012	1.080	1.215	1.350	1.620	2.025	2.430
280	0.420	0.560	0.700	0.840	0.980	1.050	1.120	1.260	1.400	1.680	2.100	2.520
290	0.435	0.580	0.725	0.870	1.015	1.087	1.160	1.305	1.450	1.740	2.175	2.610
300	0.450	0.600	0.750	0.900	1.050	1.125	1.200	1.350	1.500	1.800	2.250	2.700
310	0.465	0.620	0.775	0.930	1.085	1.162	1.240	1.395	1.550	1.860	2.325	2.790
320	0.480	0.640	0.800	0.960	1.120	1.200	1.280	1.440	1.600	1.920	2.400	2.880
330	0.495	0.660	0.825	0.990	1.155	1.237	1.320	1.485	1.650	1.980	2.475	2.970
340	0.510	0.680	0.850	1.020	1.190	1.275	1.360	1.530	1.700	2.040	2.550	3.060
350	0.525	0.700	0.875	1.050	1.225	1.312	1.400	1.575	1.750	2.100	2.625	3.150
360	0.540	0.720	0.900	1.080	1.260	1.350	1.440	1.620	1.800	2.160	2.700	3.240
370	0.555	0.740	0.925	1.110	1.295	1.387	1.480	1.665	1.850	2.220	2.775	3.330
380	0.570	0.760	0.950	1.140	1.330	1.425	1.520	1.710	1.900	2.280	2.850	3.420
390	0.585	0.780	0.975	1.170	1.365	1.462	1.560	1.755	1.950	2.340	2.925	3.510
400	0.600	0.800	1.000	1.200	1.400	1.500	1.600	1.800	2.000	2.400	3.000	3.600
425	0.637	0.850	1.062	1.275	1.487	1.595	1.700	1.912	2.125	2.550	3.187	3.825
450	0.675	0.900	1.125	1.350	1.575	1.687	1.800	2.025	2.250	2.700	3.375	4.050
475	0.712	0.950	1.187	1.425	1.662	1.782	1.900	2.137	2.375	2.850	3.562	4.275
500	0.750	1.000	1.250	1.500	1.750	1.875	2.000	2.250	2.500	3.000	3.750	4.500

For use with Table VI.

TABLE VIII

VALUES OF $[R^{1/n} - 1]$ FOR VOLUMETRIC EFFICIENCY CALCULATIONS

Values of R	Isothermal	Polytropics				
	$n = 1$	$n = 1.1$	$n = 1.2$	$n = 1.3$	$n = 1.4$	$n = 1.5$
1.4	0.400	0.358	0.324	0.295	0.272	0.251
1.6	0.600	0.533	0.480	0.435	0.399	0.368
1.8	0.800	0.706	0.632	0.572	0.522	0.480
2.0	1.000	0.878	0.782	0.704	0.641	0.587
2.2	1.200	1.048	0.929	0.834	0.756	0.691
2.4	1.400	1.216	1.074	0.961	0.869	0.793
2.6	1.600	1.384	1.217	1.085	0.979	0.891
2.8	1.800	1.550	1.358	1.208	1.086	0.987
3.0	2.000	1.715	1.498	1.328	1.192	1.080
3.2	2.200	1.879	1.636	1.447	1.295	1.172
3.4	2.400	2.042	1.773	1.563	1.397	1.261
3.6	2.600	2.204	1.908	1.679	1.497	1.349
3.8	2.800	2.366	2.042	1.792	1.595	1.435
4.0	3.000	2.526	2.175	1.905	1.692	1.520
4.5	3.500	2.925	2.502	2.180	1.928	1.726
5.0	4.000	3.319	2.824	2.449	2.157	1.924
6.0	5.000	4.098	3.451	2.968	2.596	2.302
8.0	7.000	5.622	4.657	3.951	3.416	3.000
10.0	9.000	7.111	5.813	4.878	4.179	3.642
15.0	14.000	10.728	8.552	7.029	5.919	5.082
20.0	19.000	14.232	11.139	9.018	7.498	6.368
25.0	24.000	17.658	13.620	10.894	8.966	7.550

See equations 23 and 60.

TABLE IX

Maximum Ratio of Compression as a Function of Clearance

Per Cent Clearance	Isothermal	Polytropics				
	$n = 1$	$n = 1.1$	$n = 1.2$	$n = 1.3$	$n = 1.4$	$n = 1.5$
1	101.	160.	255.	403.	636.	1020.
2	51.	75.7	113.	245.	365.	538.
3	34.3	48.8	69.5	99.	141.	200.
4	26.0	36.0	50.0	69.	96.	133.
5	21.0	28.5	38.5	52.5	71.	96.
6	17.67	23.5	31.4	41.7	52.8	74.
7	15.30	20.2	26.5	34.8	45.8	55.
8	13.50	17.5	22.7	29.5	38.3	49.6
9	12.11	15.5	20.0	25.5	32.8	42.2
10	11.00	14.0	17.8	22.6	28.7	36.5
12	9.33	11.65	14.6	18.2	22.75	28.5
14	8.14	10.05	12.4	15.25	18.90	23.3
16	7.25	8.85	10.8	13.10	16.00	19.5
18	6.55	7.90	9.55	11.50	13.90	16.8
20	6.00	7.18	8.59	10.25	12.30	14.7
25	5.00	5.88	6.90	8.10	9.50	11.2
30	4.33	5.02	5.80	6.73	7.78	9.01

From equation 24.

The table gives values of $R_m = \left(\frac{1+c}{c}\right)^n$.

TABLE X

Compression Mean Effective Pressure

Values of R	Isothermal $n = 1.0$	Polytropics $n = 1.1$	$n = 1.2$	$n = 1.3$	$n = 1.4$	$n = 1.5$
1.4	4.96	5.03	5.10	5.15	5.20	5.25
1.6	6.92	7.07	7.20	7.31	7.41	7.50
1.8	8.66	8.89	9.10	9.27	9.43	9.57
2.0	10.21	10.54	10.82	11.07	11.29	11.49
2.2	11.61	12.04	12.41	12.74	13.03	13.29
2.4	12.89	13.42	13.89	14.29	14.65	14.98
2.6	14.07	14.70	15.26	15.75	16.19	16.58
2.8	15.16	15.49	16.55	17.12	17.64	18.10
3.0	16.18	17.02	17.76	18.42	19.01	19.55
3.2	17.13	18.08	18.91	19.66	20.33	20.93
3.4	18.02	19.07	20.00	20.83	21.58	22.26
3.6	18.87	20.01	21.04	21.96	22.79	23.54
3.8	19.66	20.91	22.03	23.03	23.95	24.77
4.0	20.42	21.77	22.98	24.07	25.06	25.96
4.5	22.15	23.75	25.18	26.49	27.68	28.77
5.0	23.71	25.53	27.20	28.72	30.01	31.38
6.0	26.39	28.67	30.76	32.69	34.47	36.12
8.0	30.63	33.72	36.61	39.32	41.84	44.20
10.0	33.91	37.74	41.35	44.77	47.99	51.02
15.0	39.89	45.24	50.42	55.42	60.22	64.80
20.0	44.12	50.73	57.24	63.61	69.80	75.78
25.0	47.41	54.95	62.76	70.35	77.78	85.01

Factor for isothermal ($n = 1.0$) $= 14.73 \log_\epsilon R$.

Factor for polytropics $= \dfrac{14.73\,n}{n-1}\,[R^{(n-1)/n} - 1]$.

MEP = factor $\times$ volumetric efficiency.

Indicated horsepower $= \dfrac{MEP \times L\,A\,N}{33{,}000}$, where L is in feet and A is in square inches.

For values of $\dfrac{L\,A\,N}{33{,}000}$ see Table VI.

TABLE XI

TEMPERATURE-RISE FACTORS FOR COMPRESSION

R	Isothermal	Polytropics				
	$n = 1$	$n = 1.1$	$n = 1.2$	$n = 1.3$	$n = 1.4$	$n = 1.5$
1.4	1.000	1.031	1.058	1.081	1.101	1.119
1.6	1.000	1.044	1.081	1.112	1.144	1.170
1.8	1.000	1.055	1.103	1.145	1.183	1.216
2.0	1.000	1.065	1.122	1.173	1.219	1.260
2.2	1.000	1.074	1.140	1.200	1.253	1.301
2.4	1.000	1.083	1.157	1.224	1.284	1.339
2.6	1.000	1.091	1.173	1.247	1.314	1.375
2.8	1.000	1.096	1.187	1.268	1.342	1.409
3.0	1.000	1.105	1.201	1.289	1.369	1.442
3.2	1.000	1.111	1.214	1.308	1.394	1.474
3.4	1.000	1.118	1.226	1.326	1.419	1.504
3.6	1.000	1.123	1.238	1.344	1.442	1.533
3.8	1.000	1.129	1.249	1.361	1.464	1.560
4.0	1.000	1.134	1.260	1.377	1.486	1.587
4.5	1.000	1.146	1.285	1.415	1.537	1.651
5.0	1.000	1.158	1.308	1.450	1.582	1.710
6.0	1.000	1.177	1.348	1.512	1.668	1.817
8.0	1.000	1.208	1.414	1.616	1.811	2.000
10.0	1.000	1.233	1.468	1.701	1.931	2.154
15.0	1.000	1.279	1.570	1.868	2.168	2.466
20.0	1.000	1.313	1.648	1.996	2.353	2.714
25.0	1.000	1.339	1.710	2.102	2.508	2.923

See equation 29.

$$\text{Factor} = \frac{T_2}{T_1} = R^{(n-1)/n}$$

TABLE XII, *Part I*

Volumetric Efficiency of Compression

R	Clearance in Per Cent 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
1.3	97.7	97.5	97.3	97.0	96.8	96.5	96.3	96.0	95.8	95.6	95.3	95.1	94.8	94.6	94.3	94.1	93.8	93.6	93.4	93.1
1.4	97.7	97.3	97.0	96.7	96.4	96.0	95.7	95.4	95.1	94.7	94.4	94.1	93.8	93.5	93.1	92.8	92.5	92.2	91.8	91.5
1.5	97.6	97.2	96.8	96.4	96.0	95.6	95.2	94.8	94.4	94.0	93.6	93.2	92.8	92.4	92.0	91.6	91.2	90.8	90.4	90.0
1.6	97.5	97.0	96.6	96.1	95.6	95.1	94.6	94.2	93.7	93.2	92.7	92.2	91.8	91.3	90.8	90.3	89.8	89.4	88.9	88.4
1.8	97.4	96.7	96.1	95.5	94.8	94.2	93.6	92.9	92.3	91.7	91.0	90.4	89.8	89.2	88.5	87.9	87.3	86.6	86.0	85.4
2.0	97.2	96.4	95.6	94.9	94.1	93.3	92.5	91.7	91.0	90.1	89.4	88.6	87.8	87.0	86.3	85.5	84.7	83.9	83.1	82.4
2.2	97.0	96.1	95.2	94.3	93.3	92.4	91.5	90.6	89.6	88.7	87.8	86.9	85.9	85.0	84.1	83.1	82.2	81.3	80.3	79.4
2.4	96.9	95.8	94.8	93.7	92.6	91.6	90.5	89.4	88.3	87.3	86.2	85.1	84.0	83.0	81.9	80.8	79.7	78.7	77.6	76.5
2.6	96.8	95.6	94.3	93.1	91.9	90.7	89.5	88.3	87.0	85.8	84.6	83.4	82.2	81.0	79.7	78.5	77.3	76.1	74.9	73.7
2.8	96.6	95.3	93.9	92.6	91.2	89.8	88.5	87.1	85.8	84.4	83.1	81.7	80.3	79.0	77.6	76.3	74.9	73.6	72.2	70.8
3.0	96.5	95.0	93.5	92.0	90.5	89.0	87.5	86.0	84.5	83.0	81.5	80.0	78.5	77.0	75.5	74.0	72.5	71.0	69.5	68.0
3.2	96.4	94.7	93.1	91.5	89.8	88.2	86.5	84.9	83.3	81.6	80.0	78.4	76.7	75.1	73.5	71.8	70.2	68.6	66.9	65.3
3.4	96.2	94.4	92.7	90.9	89.1	87.4	85.6	83.8	82.0	80.3	78.5	76.6	74.9	73.2	71.4	69.6	67.8	66.1	64.3	62.5
3.6	96.1	94.2	92.3	90.4	88.5	86.6	84.6	82.7	80.8	78.9	77.0	75.1	73.2	71.3	69.4	67.5	65.6	63.7	61.7	59.8
3.8	96.0	93.9	91.9	89.8	87.8	85.7	83.7	81.7	79.6	77.6	75.5	73.5	71.4	69.4	67.4	65.3	63.3	61.2	59.2	57.2
4.0	95.8	93.6	91.5	89.3	87.1	84.9	82.8	80.6	78.4	76.2	74.1	71.9	69.7	67.6	65.4	63.2	61.0	58.8	56.7	54.5
4.5	95.5	93.0	90.5	88.0	85.5	83.0	80.5	78.0	75.5	73.0	70.5	68.0	65.5	63.0	60.5	58.0	55.5	53.0	50.5	48.0
5.0	95.2	92.3	89.6	86.7	83.9	81.1	78.2	75.4	72.6	69.8	66.9	64.1	61.3	58.5	55.6	52.8	50.0	47.2	44.3	41.6
6.0	94.5	91.1	87.6	84.2	80.7	77.3	73.8	70.4	66.9	63.5	70.0	56.6	53.1	49.7	46.2	42.8	39.3	35.9	32.4	29.0
7.0	93.8	89.7	85.5	81.4	77.2	73.0	68.9	64.7	60.6	56.4	52.2	48.1	43.9	39.8	35.6	31.4	27.3	23.1	19.0	14.8
8.0	93.3	88.7	84.0	79.4	74.7	70.0	65.4	60.7	56.1	51.4	46.8	42.1	37.5	32.8	28.1	23.5	18.8	14.2	9.5	4.9
9.0	92.8	87.5	82.3	77.0	71.8	66.6	61.3	56.1	50.8	45.6	40.4	35.1	29.9	24.6	19.4	14.2	8.9			
10.0	92.2	86.4	80.6	74.7	68.9	63.1	57.3	51.5	45.7	39.9	34.1	28.2	22.4	16.6	10.8	5.0				
12.0	91.2	84.3	77.5	70.7	63.8	57.0	50.2	43.4	36.6	29.7	22.9	16.0	9.2	2.4						
14.0	90.0	82.0	73.9	65.9	57.9	49.9	41.9	33.8	25.8	17.8	9.8	1.8								
15.0	89.4	80.9	72.3	63.8	55.2	46.7	38.1	29.6	21.0	12.5	3.9									
16.0	88.9	79.8	70.8	61.7	52.6	43.6	34.4	25.4	16.3	7.2										
18.0	87.9	77.8	67.7	57.6	47.5	37.4	27.3	17.2	7.1											
20.0	86.9	75.7	64.6	53.4	42.3	31.2	20.0	8.9												
25.0	84.4	70.8	57.1	43.5	29.9	16.3	2.7													

See equation 60. **Based on n = 1.2.**

TABLE XII, *Part II*

Volumetric Efficiency of Compression

R	Clearance in Per Cent																	
	20	21	22	23	24	25	26	27	28	29	30	32	34	36	38	40	45	50
1.3	93.1	92.9	92.6	92.4	92.1	91.9	91.6	91.4	91.2	90.9	90.7	90.2	89.7	89.2	88.7	88.2	87.0	85.8
1.4	91.5	91.2	90.9	90.5	90.2	89.9	89.6	89.2	88.9	88.6	88.3	87.6	87.0	86.3	85.7	85.0	83.4	81.8
1.5	90.0	89.6	89.2	88.8	88.4	88.0	87.5	87.1	86.7	86.3	85.9	85.1	84.3	83.5	82.7	81.9	79.9	77.9
1.6	88.4	87.9	87.4	87.0	86.5	86.0	85.5	85.0	84.6	84.1	83.6	82.6	81.7	80.7	79.8	78.8	76.4	74.0
1.8	85.4	84.7	84.1	83.5	82.8	82.3	81.6	80.9	80.3	79.7	79.0	77.8	76.5	75.2	74.0	72.7	69.6	66.4
2.0	82.4	81.6	80.8	80.0	79.2	78.4	77.7	76.9	76.1	75.3	74.5	73.0	71.4	69.8	68.3	66.7	62.8	58.9
2.2	79.4	78.5	77.6	76.6	75.7	74.7	73.8	72.9	72.0	71.1	70.1	68.3	66.4	64.6	62.7	60.8	56.2	51.6
2.4	76.5	75.4	74.4	73.3	72.2	71.1	70.1	69.0	67.9	66.8	65.8	63.6	61.5	59.3	57.2	55.0	49.7	44.3
2.6	73.7	72.4	71.2	70.0	68.8	67.6	66.4	65.1	63.9	62.7	61.5	59.1	56.6	54.2	51.7	49.3	43.2	37.1
2.8	70.8	69.5	68.1	66.8	65.4	64.0	62.7	61.3	60.0	58.6	57.3	54.5	51.8	49.1	46.4	43.7	36.9	30.1
3.0	68.0	66.5	65.0	63.6	62.0	60.6	59.1	57.6	56.1	54.6	53.1	50.1	47.1	44.1	41.1	38.1	30.6	23.1
3.2	65.3	63.6	62.0	60.4	58.7	57.1	55.5	53.8	52.2	50.6	48.9	45.6	42.4	39.1	35.8	32.6	24.4	16.2
3.4	62.5	60.8	59.0	57.2	55.4	53.7	51.9	50.1	48.4	46.6	44.8	41.2	37.7	34.2	30.6	27.1	18.2	9.3
3.6	59.8	57.9	56.0	54.1	52.2	50.3	48.4	46.5	44.6	42.7	40.8	36.9	33.1	29.3	25.5	21.7	12.1	2.6
3.8	57.2	55.1	53.1	51.0	49.0	46.9	44.9	42.9	40.8	38.8	36.7	32.6	28.6	24.5	20.4	16.3	6.1	
4.0	54.5	52.3	50.1	48.0	45.8	43.6	41.4	39.3	37.1	34.9	32.7	28.4	24.0	19.7	15.3	11.0	0.1	
4.5	48.0	45.5	43.0	40.5	38.0	35.4	32.9	30.4	27.9	25.4	22.9	17.9	12.9	7.9	2.9			
5.0	41.6	38.7	35.9	33.0	30.2	27.4	24.6	21.8	18.9	16.1	13.3	7.6	2.0					
6.0	29.0	25.5	22.1	18.6	15.2	11.7	8.3	4.8	1.4									
7.0	14.8	10.6	6.5	2.3														
8.0	4.9	2.0																

Based on equation 60:

$$E_v = 0.98 - c\,(R^{1/n} - 1)$$

where E_v = volumetric efficiency.

R = ratio of compression.

n = exponent of compression. (Value of 1.2 used in table.)

TABLE XIII

Brake Horsepower per Million Cubic Feet per 24 Hours

Values of R	Isothermal	Polytropics			
	$n = 1.0$	$n = 1.1$	$n = 1.2$	$n = 1.3$	$n = 1.4$
1.0	10.0	10.0	10.0	10.0	10.0
1.4	25.1	25.3	25.5	25.7	25.8
1.6	31.1	31.6	32.0	32.3	32.6
1.8	36.5	37.2	37.9	38.4	38.9
2.0	41.4	42.4	43.3	44.0	44.7
2.2	45.7	47.1	48.2	49.3	50.2
2.4	49.9	51.4	52.9	54.2	55.3
2.6	53.6	55.6	57.3	58.8	60.1
2.8	57.1	58.2	61.4	63.3	64.8
3.0	60.5	63.2	65.4	67.5	69.3
3.2	63.7	66.7	69.3	71.6	73.7
3.4	66.7	70.0	72.9	75.5	77.9
3.6	69.7	73.2	76.4	79.3	81.9
3.8	72.5	76.3	79.8	83.1	85.9
4.0	75.1	79.4	83.2	86.7	89.8
4.5	81.5	86.6	91.3	95.5	99.3
5.0	87.6	93.6	99.0	103.9	108.1
6.0	99.2	106.8	113.8	120.2	126.2
8.0	121.6	132.7	143.0	152.7	161.8
10.0	145.0	159.7	173.8	187.2	199.8

This table is based on theoretical horsepower values given in Table V.

$$BHP \text{ per } MMcf = (\text{Theoretical } HP + 10)\left(1 + \frac{R^2}{350}\right) \qquad (62)$$

To correct for gas law deviation, multiply by value of Y from equation 75.

TABLE XIV

Compressibility Factors for Air

Gauge Pressure	Temperature, °F.						
	60	80	100	150	200	300	400
0	1.000	1.000	1.000	1.000	1.001	1.001	1.001
100	0.997	0.998	0.999	1.000	1.001	1.002	1.003
200	0.995	0.996	0.998	1.000	1.002	1.004	1.005
300	0.993	0.995	0.997	1.000	1.003	1.006	1.007
400	0.991	0.993	0.996	1.001	1.004	1.008	1.009
500	0.989	0.992	0.995	1.001	1.006	1.010	1.012
600	0.987	0.992	0.995	1.002	1.007	1.012	1.014
700	0.986	0.991	0.995	1.003	1.009	1.014	1.017
800	0.985	0.990	0.995	1.004	1.010	1.018	1.020
900	0.984	0.990	0.995	1.005	1.012	1.019	1.022
1000	0.983	0.990	0.995	1.006	1.013	1.021	1.025
1100	0.982	0.989	0.996	1.008	1.015	1.024	1.028
1200	0.982	0.989	0.997	1.009	1.017	1.027	1.031
1300	0.982	0.990	0.998	1.011	1.020	1.029	1.034
1400	0.983	0.992	1.000	1.014	1.022	1.032	1.037
1500	0.985	0.994	1.003	1.017	1.026	1.035	1.040

TABLE XV

COMPRESSIBILITY FACTORS FOR METHANE
Specific Gravity 0.553

Gauge Pressure	Temperature, °F.						
	60	80	100	150	200	300	400
0	1.000	1.000	1.000	1.000	1.000	1.000	1.000
100	0.987	0.989	0.990	0.993	0.995	0.998	0.999
200	0.974	0.978	0.981	0.987	0.991	0.996	0.998
300	0.961	0.967	0.971	0.980	0.986	0.994	0.998
400	0.948	0.956	0.962	0.973	0.982	0.993	0.997
500	0.936	0.945	0.953	0.967	0.978	0.991	0.997
600	0.923	0.934	0.943	0.960	0.974	0.990	0.998
700	0.910	0.922	0.934	0.954	0.969	0.988	0.998
800	0.897	0.912	0.926	0.948	0.965	0.987	0.998
900	0.884	0.902	0.918	0.942	0.961	0.985	0.998
1000	0.872	0.892	0.910	0.937	0.957	0.983	0.998
1100	0.860	0.882	0.902	0.931	0.953	0.982	0.998
1200	0.850	0.874	0.894	0.926	0.951	0.981	0.998
1300	0.840	0.865	0.887	0.922	0.948	0.981	0.998
1400	0.830	0.857	0.880	0.918	0.946	0.980	0.998
1500	0.822	0.849	0.874	0.914	0.944	0.980	0.999
1600	0.813	0.842	0.868	0.910	0.942	0.980	1.000
1700	0.806	0.835	0.862	0.908	0.941	0.980	1.001
1800	0.799	0.830	0.858	0.905	0.940	0.980	1.002
1900	0.793	0.824	0.853	0.903	0.939	0.980	1.003
2000	0.788	0.820	0.849	0.902	0.938	0.980	1.004
2100	0.783	0.817	0.846	0.901	0.937	0.981	1.006
2200	0.781	0.814	0.844	0.900	0.937	0.982	1.007
2300	0.779	0.812	0.843	0.900	0.936	0.983	1.009
2400	0.778	0.811	0.842	0.900	0.937	0.984	1.010
2500	0.778	0.811	0.842	0.900	0.938	0.985	1.012
2600	0.778	0.812	0.842	0.900	0.939	0.986	1.014
2700	0.778	0.813	0.843	0.900	0.940	0.988	1.017
2800	0.780	0.815	0.844	0.901	0.942	0.990	1.019
2900	0.782	9.818	0.846	0.903	0.943	0.992	1.021
3000	0.783	0.820	0.848	0.904	0.945	0.994	1.023
3100	0.786	0.823	0.850	0.905	0.947	0.996	1.025
3200	0.789	0.825	0.852	0.907	0.949	0.998	1.028
3300	0.792	0.828	0.855	0.909	0.951	1.000	1.030
3400	0.796	0.831	0.857	0.910	0.953	1.003	1.033
3500	0.800	0.835	0.860	0.912	0.955	1.005	1.035
3600	0.802	0.837	0.863	0.914	0.957	1.008	1.038

TABLE XVI

COMPRESSIBILITY FACTORS FOR NATURAL GAS

(Free of Air and CO_2)

	°F.	Gauge Pressure, pounds per square inch							
		100	200	300	400	500	600	700	800
Sp. Gr. 0.60	60	0.987	0.971	0.955	0.939	0.924	0.910	0.896	0.882
	80	0.988	0.974	0.961	0.947	0.934	0.920	0.909	0.898
	100	0.990	0.977	0.968	0.954	0.943	0.931	0.920	0.910
	150	0.992	0.983	0.975	0.966	0.958	0.951	0.943	0.935
	200	0.994	0.987	0.981	0.975	0.969	0.963	0.957	0.951
	300	0.997	0.994	0.989	0.985	0.982	0.978	0.974	0.971
	400	0.998	0.995	0.993	0.991	0.989	0.986	0.984	0.982
Sp. Gr. 0.70	60	0.980	0.957	0.935	0.910	0.885	0.864	0.842	0.820
	80	0.982	0.962	0.942	0.921	0.900	0.880	0.861	0.841
	100	0.984	0.967	0.949	0.930	0.912	0.894	0.877	0.860
	150	0.988	0.976	0.962	0.948	0.937	0.923	0.911	0.898
	200	0.991	0.982	0.972	0.962	0.952	0.943	0.932	0.926
	300	0.995	0.989	0.983	0.977	0.972	0.966	0.960	0.954
	400	0.997	0.993	0.989	0.986	0.982	0.978	0.975	0.971
Sp. Gr. 0.80	60	0.973	0.942	0.910	0.877	0.844	0.815	0.786	0.758
	80	0.976	0.949	0.921	0.891	0.862	0.840	0.814	0.790
	100	0.979	0.955	0.929	0.903	0.879	0.860	0.838	0.817
	150	0.984	0.966	0.947	0.928	0.910	0.892	0.875	0.857
	200	0.988	0.975	0.960	0.946	0.932	0.917	0.904	0.890
	300	0.993	0.985	0.976	0.967	0.959	0.951	0.942	0.933
	400	0.995	0.991	0.985	0.980	0.974	0.969	0.963	0.957
Sp. Gr. 0.90	60	0.965	0.924	0.883	0.840	0.800			
	80	0.969	0.933	0.897	0.862	0.828			
	100	0.972	0.942	0.912	0.885	0.857			
	150	0.978	0.957	0.937	0.917	0.899			
	200	0.984	0.968	0.953	0.937	0.923			
	300	0.990	0.981	0.972	0.963	0.954			
	400	0.994	0.988	0.982	0.976	0.970			
Sp. Gr. 1.00	60	0.955	0.902	0.850	0.800	0.752			
	80	0.961	0.915	0.872	0.830	0.788			
	100	0.964	0.926	0.889	0.852	0.818			
	150	0.973	0.946	0.921	0.896	0.873			
	200	0.980	0.960	0.940	0.922	0.903			
	300	0.988	0.976	0.964	0.953	0.941			
	400	0.992	0.985	0.977	0.970	0.962			
Sp. Gr. 1.20	60	0.931	0.851	0.775	0.709				
	80	0.940	0.869	0.805	0.745				
	100	0.947	0.886	0.827	0.772				
	150	0.961	0.918	0.876	0.835				
	200	0.971	0.938	0.905	0.873				
	300	0.983	0.962	0.943	0.922				
	400	0.989	0.976	0.963	0.950				
Sp. Gr. 1.40	60	0.899	0.787	0.694					
	80	0.911	0.816	0.733					
	100	0.923	0.837	0.757					
	150	0.945	0.883	0.826					
	200	0.959	0.911	0.865					
	300	0.975	0.946	0.917					
	400	0.984	0.966	0.947					

TABLE XVII, *Part I*

COMPRESSOR CYLINDER DATA

Cylinder Diameter in Inches	Manufacturer	Stroke in Inches	Revolutions per Minute	Rod Diameter in Inches	Maximum Working Pressure in Pounds per Square Inch	Approximate Valve Area in Square Inches	Average Clearance in Per Cent
3.00	B	12	350	1.75	1800	0.60	15.6
3.25	B	12	350	1.75	1800	0.60	12.3
4.25	A	10	275	1.75	500	1.00	12.3
4.50	A	10	275	1.75	500	1.00	10.8
4.50	B	12	350	1.75	1000	1.50	12.0
4.75	B	12	350	1.75	1000	1.50	11.2
5.00	B	12	350	1.75	1000	1.50	14.0
5.00	A	10	275	1.75	1000	1.50	8.8
5.25	A	10	275	1.75	1000	1.50	7.3
5.25	B	12	350	1.75	1000	1.50	12.7
5.50	C	20	200	2.25	1000	8.00	7.5
5.50	D	20	200	2.25	1000	6.50	10.6
5.75	C	20	200	2.25	1000	8.00	9.0
5.75	D	20	200	2.25	1000	6.50	9.7
6.00	A	10	275	1.75	1000	1.50	5.8
6.00	B	12	350	1.75	800	2.70	10.6
6.25	A	10	275	1.75	1000	1.50	5.4
6.25	B	12	350	1.75	800	2.70	10.0
6.25	C	14	300	2.25	900	4.00	12.5
6.50	C	14	300	2.25	900	4.00	11.5
7.00	A	10	275	1.75	350	1.75	8.0
7.25	A	10	275	1.75	350	1.75	7.5
7.25	C	20	200	2.25	800	8.00	7.4
7.50	C	20	200	2.25	800	8.00	6.9
7.50	C	20	200	2.25	500	8.00	6.9
7.75	C	20	200	2.25	500	8.00	6.5
8.00	B	12	350	1.75	600	6.70	12.5
8.25	B	12	350	1.75	600	6.70	12.8
8.50	D	20	200	2.25	500	6.50	8.0
8.75	D	20	200	2.25	500	6.50	7.8
9.50	E	20	185	3.00	1000	6.90	11.3
9.75	E	20	185	3.00	1000	6.90	10.6
10.00	A	10	275	1.75	150	2.80	6.5
10.00	B	12	350	1.75	250	10.00	9.8
10.25	A	10	275	1.75	150	2.80	6.2
10.25	B	12	350	1.75	250	10.00	9.4
10.50	D	20	200	2.25	375	10.30	16.8
10.50	D	14	300	2.25	375	10.30	16.7
10.75	D	20	200	2.25	375	10.30	15.9
10.75	D	14	300	2.25	375	10.30	16.0
12.00	A	10	275	1.75	100	4.15	7.0
12.00	B	12	350	1.75	150	13.80	10.0
12.00	C	20	200	2.25	200	7.10	4.6
12.25	A	10	275	1.75	100	4.15	6.8
12.25	B	12	350	1.75	150	13.80	9.6
12.25	C	20	200	2.25	200	7.10	4.4
12.50	C	20	200	2.25	125	14.20	7.9
12.75	C	20	200	2.25	125	14.20	7.8

TABLE XVII, *Part II*

COMPRESSOR CYLINDER DATA

Cylinder Diameter in Inches	Manufacturer	Stroke in Inches	Revolutions per Minute	Rod Diameter in Inches	Maximum Working Pressure in Pounds per Square Inch	Approximate Valve Area in Square Inches	Average Clearance in Per Cent
13.00	C	20	200	2.25	125	14.20	6.5
13.00	D	20	200	2.25	125	20.60	11.2
13.00	E	20	185	3.00	500	6.90	13.8
13.25	C	20	200	2.25	125	14.70	6.4
13.25	D	20	200	2.25	125	20.60	10.8
13.25	E	20	185	3.00	500	6.90	13.8
15.50	B	12	350	1.75	125	21.25	9.0
15.75	B	12	350	1.75	125	21.25	8.8
17.00	A	10	275	1.75	50	5.25	6.5
17.00	F	36	120	3.00	500	6.90	8.0
17.25	A	10	275	1.75	50	5.25	6.6
17.25	F	36	120	3.00	500	6.90	8.0
17.50	C	20	200	2.25	75	14.20	4.0
17.50	D	20	200	2.50	100	29.80	9.5
17.75	C	20	200	2.25	75	14.20	4.0
17.75	D	20	200	2.50	100	29.80	9.3
18.00	C	20	200	2.25	75	14.20	3.9
18.00	D	20	200	2.50	100	29.80	8.8
18.25	C	20	200	2.25	75	14.20	3.8
18.25	D	20	200	2.50	100	29.80	8.5
19.00	D	20	200	2.50	100	29.80	11.9
19.00	D	14	300	2.25	100	29.80	11.4
19.25	D	20	200	2.50	100	29.80	11.6
19.25	D	14	300	2.25	100	29.80	10.6
22.00	E	20	185	3.00	150	22.40	7.3
22.25	E	20	185	3.00	150	22.40	7.4
31.00	F	36	120	3.00	50	166.00	11.6
31.25	F	36	120	3.00	50	166.00	11.5
33.00	F	36	120	3.00	50	182.00	10.4
33.25	F	36	120	3.00	50	182.00	10.3

TABLE XVIII, *Part I*

Data for Solution of Weymouth's Equation for Pipe-Line Pressure-Drop

Values of $\sqrt{1/L}$

Feet		Feet		Feet		Feet		Feet		Feet	
100	0.1000	650	0.0392	1500	0.0258	2600	0.0196	3700	0.0164	6,500	0.0124
150	0.0816	700	0.0378	1600	0.0250	2700	0.0192	3800	0.0162	7,000	0.0120
200	0.0707	750	0.0365	1700	0.0243	2800	0.0189	3900	0.0160	7,500	0.0116
250	0.0633	800	0.0354	1800	0.0236	2900	0.0186	4000	0.0158	8,000	0.0112
300	0.0578	850	0.0343	1900	0.0229	3000	0.0183	4200	0.0154	9,000	0.0105
350	0.0534	900	0.0333	2000	0.0224	3100	0.0180	4400	0.0151	10,000	0.0100
400	0.0500	1000	0.0316	2100	0.0218	3200	0.0177	4600	0.0147	15,000	0.0082
450	0.0471	1100	0.0301	2200	0.0213	3300	0.0174	4800	0.0144	20,000	0.0071
500	0.0447	1200	0.0289	2300	0.0208	3400	0.0172	5000	0.0141	25,000	0.0063
550	0.0427	1300	0.0277	2400	0.0204	3500	0.0169	5500	0.0135	30,000	0.0052
600	0.0408	1400	0.0267	2500	0.0200	3600	0.0167	6000	0.0129	50,000	0.0045

Values of $48.76\sqrt{1/G}$

G		G		G		G		G	
0.50	68.99	0.65	60.46	0.80	54.51	0.95	50.03	1.20	44.52
0.55	65.73	0.70	58.27	0.85	52.90	1.00	48.76	1.30	42.76
0.60	62.95	0.75	56.32	0.90	51.39	1.10	46.52	1.40	41.20

Values of $d^{8/3}$ for Nominal Pipe Sizes

1″	1.14	2″	6.93	4″	41.02	10″	488.31	16″ O.D.	1430.16
1¼″	2.36	2½″	11.14	6″	122.33	12″	769.97	18″ O.D.	1965.75
1½″	3.56	3″	19.87	8″	262.10	14″	983.05	20″ O.D.	2636.64

Values of $d^{8/3}$ by Tenths of an Inch

d	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
1.0	1.0	1.3	1.6	2.0	2.4	2.9	3.5	4.1	4.8	5.5
2.0	6.3	7.2	8.2	9.2	10.3	11.5	12.8	14.1	15.6	17.1
3.0	18.7	20.4	22.2	24.1	26.1	28.2	30.4	32.7	35.2	37.7
4.0	40.3	43.1	45.9	48.9	52.0	55.2	58.5	62.0	65.6	69.3
5.0	73.1	77.1	81.2	85.4	89.8	94.3	98.9	103.7	108.6	113.7
6.0	118.9	124.2	129.9	135.4	141.2	147.1	153.3	159.5	166.0	172.6
7.0	179.3	186.2	193.3	200.5	207.9	215.5	223.3	231.2	239.3	247.6
8.0	256.0	264.6	273.4	282.4	291.6	300.9	310.5	320.2	330.1	340.2
9.0	350.5	361.0	371.6	382.5	393.6	404.8	416.3	428.0	439.8	451.9
10.0	464.2	476.6	489.3	502.2	515.3	528.7	542.2	555.9	569.9	584.1
11.0	598.5	613.1	627.9	643.0	658.3	673.8	689.5	705.5	721.7	738.1
12.0	754.8	771.7	788.8	806.1	823.7	841.6	859.7	878.0	896.5	915.3
13.0	934.4	953.7	973.2	993.0	1013.	1033.	1054.	1075.	1096.	1117.
14.0	1138.	1160.	1182.	1205.	1227.	1250.	1273.	1297.	1321.	1344.
15.0	1368.	1393.	1418.	1443.	1468.	1494.	1519.	1545.	1572.	1599.
16.0	1625.	1653.	1680.	1708.	1736.	1765.	1793.	1822.	1851.	1881.
17.0	1911.	1941.	1971.	2002.	2033	2064.	2096.	2128.	2160.	2193.
18.0	2225.	2259.	2292.	2326.	2360.	2394.	2429.	2464.	2499.	2535.
19.0	2570.	2607.	2643.	2680.	2717.	2755.	2793.	2831.	2869.	2908.
20.0	2947.	2987.	3026.	3067.	3107.	3148.	3196.	3230.	3272.	3314.
21.0	3357.	3400.	3443.	3486.	3530.	3574.	3619.	3663.	3709.	3754.
22.0	3800.	3846.	3893.	3940.	3987.	4035.	4083.	4131.	4180.	4229.
23.0	4278.	4328.	4378.	4429.	4480.	4531.	4582.	4634.	4687.	4739.
24.0	4792.	4846.	4900.	4954.	5008.	5063.	5119.	5174.	5230.	5287.
25.0	5344.	5401.	5458.	5516.	5575.	5633.	5693.	5752.	5812.	5872.

See equation 91.

TABLE XVIII, *Part II*

Data for Solution of Weymouth's Equation for Pipe-Line Pressure Drop

Values of $\sqrt{U(2 - U)}$

U	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.0	.....	0.045	0.063	0.077	0.089	0.100	0.109	0.118	0.126	0.234
1.0	0.141	0.147	0.155	0.161	0.167	0.172	0.178	0.183	0.189	0.194
2.0	0.199	0.204	0.209	0.213	0.218	0.222	0.226	0.231	0.235	0.239
3.0	0.243	0.247	0.251	0.255	0.258	0.262	0.266	0.270	0.273	0.277

U	0	1	2	3	4	5	6	7	8	9
0	.....	0.141	0.199	0.243	0.280	0.312	0.341	0.367	0.392	0.415
10	0.436	0.456	0.475	0.493	0.510	0.527	0.542	0.558	0.572	0.586
20	0.600	0.613	0.626	0.638	0.650	0.661	0.672	0.683	0.694	0.704
30	0.714	0.724	0.733	0.742	0.751	0.760	0.768	0.776	0.785	0.792
40	0.800	0.807	0.814	0.821	0.828	0.835	0.842	0.848	0.854	0.860
50	0.866	0.872	0.877	0.882	0.889	0.893	0.897	0.902	0.907	0.911
60	0.916	0.920	0.925	0.929	0.932	0.936	0.940	0.943	0.947	0.950

Pipe Dimensions

Pipe Size	Outside Diameter	Inside Diameter		
		Standard	Extra Heavy	Double Extra Heavy
1	1.315	1.049	0.957	0.599
1¼	1.660	1.380	1.278	0.896
1½	1.900	1.610	1.500	1.100
2	2.375	2.067	1.939	1.503
2½	2.875	2.469	2.323	1.771
3	3.500	3.068	2.900	2.300
4	4.500	4.026	3.826	3.152
6	6.625	6.065	5.761	4.897
8	8.625	8.071	7.625	6.875
10	10.750	10.192	9.750	
12	12.750	12.090	11.750	
14 O.D.	14.000	13.250		
16 O.D.	16.000	15.250		
18 O.D.	18.000	17.182		
20 O.D.	20.000	19.182		

See equation 91.

TABLE XIX

Final Pressures for Percentage Pipe-Line Pressure Drop

Initial Pressure		Percentage of Pressure Drop in Pipe Line													
		1	2	3	4	5	6	8	10	15	20	25	30	40	50
Pounds per Square Inch Gauge	30	29.6	29.1	28.7	28.2	27.8	27.3	26.4	25.5	23.3	21.0	18.8	1.66	12.1	7.6
	29	28.6	28.1	27.7	27.2	26.8	26.4	25.5	24.6	22.4	20.2	18.1	15.9	11.5	7.1
	28	27.6	27.1	26.7	26.3	25.9	25.4	24.6	23.7	21.6	19.4	17.3	15.2	10.9	6.6
	27	26.6	26.2	25.7	25.3	23.9	24.5	23.7	22.8	20.7	18.6	16.6	14.5	10.3	6.1
	26	25.6	25.2	24.8	24.4	23.9	23.6	22.7	21.9	19.9	17.8	15.8	13.8	9.7	5.6
	25	24.6	24.2	23.8	23.4	23.0	22.6	21.8	21.0	19.0	17.0	15.0	13.1	9.1	5.1
	24	23.6	23.2	22.8	22.4	22.1	21.7	20.9	20.1	18.2	16.2	14.3	12.4	8.5	4.6
	23	22.6	22.2	21.9	21.5	21.1	20.7	20.0	19.2	17.3	15.4	13.6	11.7	7.9	4.1
	22	21.6	21.3	20.9	20.5	20.2	19.8	19.1	18.3	16.5	14.6	12.8	11.0	7.3	3.6
	21	20.6	20.3	19.9	19.6	19.2	18.9	18.1	17.4	15.6	13.8	12.1	10.3	6.7	3.1
	20	19.6	19.3	19.0	18.6	18.3	17.9	17.2	16.5	14.8	13.0	11.3	9.6	6.1	2.6
	19	18.7	18.3	18.0	17.6	17.3	17.0	16.3	15.6	13.9	12.2	10.6	8.9	5.5	2.1
	18	17.7	17.3	17.0	16.7	16.4	16.0	15.4	14.7	13.1	11.4	9.8	8.2	4.9	1.6
	17	16.7	16.4	16.0	15.7	15.4	15.1	14.5	13.8	12.2	10.6	9.1	7.5	4.3	1.1
	16	15.7	15.4	15.1	14.8	14.5	14.2	13.5	12.9	11.4	9.8	8.3	6.8	3.7	0.6
	15	14.7	14.4	14.1	13.8	13.5	13.2	12.6	12.0	10.5	9.0	7.6	6.1	3.1	0.1
	14	13.7	13.4	13.1	12.8	12.6	12.3	11.7	11.1	9.7	8.2	6.8	5.4	2.5	0.7
	13	12.7	12.4	12.2	11.9	11.6	11.3	10.8	10.2	8.8	7.4	6.1	4.7	1.9	1.8
	12	11.7	11.5	11.2	10.9	10.7	10.4	9.9	9.3	8.0	6.6	5.3	4.0	1.3	2.8
	11	10.7	10.5	10.2	10.0	9.7	9.5	8.9	8.4	7.1	5.8	4.6	3.3	0.7	3.8
	10	9.7	9.5	9.3	9.0	8.8	8.5	8.0	7.5	6.3	5.0	3.8	2.6	0.1	4.8
	9	8.8	8.5	8.3	8.0	7.8	7.6	7.1	6.6	5.4	4.2	3.1	1.9	1.0	5.8
	8	7.8	7.6	7.3	7.1	6.9	6.6	6.2	5.7	4.6	3.4	2.3	1.2	2.2	6.8
	7	6.8	6.6	6.3	6.1	5.9	5.7	5.3	4.8	3.7	2.6	1.6	0.5	3.4	7.8
	6	5.8	5.6	5.4	5.2	5.0	4.8	4.3	3.9	2.9	1.8	0.8	0.4	4.7	8.9
	5	4.8	4.6	4.4	4.2	4.0	3.8	3.4	3.0	2.0	1.0	0.1	1.9	5.9	9.9
	4	3.8	3.6	3.4	3.2	3.0	2.9	2.5	2.1	1.2	0.2	1.4	3.3	7.1	10.9
	3	2.8	2.6	2.5	2.3	2.1	1.9	1.6	1.2	0.3	1.1	2.9	4.7	8.3	11.9
	2	1.8	1.7	1.5	1.3	1.2	1.0	0.7	0.3	1.0	2.7	4.4	6.1	9.6	13.0
	1	0.8	0.7	0.5	0.4	0.2	0.1	0.5	1.2	2.8	4.4	6.0	7.6	10.8	14.0
	0	0.3	0.6	0.9	1.2	1.5	1.8	2.4	3.0	4.5	6.0	7.5	9.0	12.0	15.0
Inches of Mercury Vacuum	1	1.3	1.6	1.9	2.2	2.4	2.7	3.3	3.9	5.3	6.8	8.2	9.7	12.6	15.5
	2	2.3	2.6	2.8	3.1	3.4	3.7	4.2	4.8	6.2	7.6	9.0	10.4	13.2	16.0
	3	3.3	3.5	3.8	4.1	4.3	4.6	5.2	5.7	7.0	8.4	9.7	11.1	13.8	16.5
	4	4.3	4.5	4.8	5.0	5.3	5.6	6.1	6.6	7.9	9.2	10.5	11.8	14.4	17.0
	5	5.2	5.5	5.7	6.0	6.2	6.5	7.0	7.5	8.7	10.0	11.2	12.5	15.0	17.5
	6	6.2	6.5	6.7	7.0	7.2	7.4	7.9	8.4	9.6	10.8	12.0	13.2	15.6	18.0
	7	7.2	7.5	7.7	7.9	8.1	8.4	8.8	9.3	10.4	11.6	12.7	13.9	16.2	18.5
	8	8.2	8.4	8.7	8.9	9.1	9.3	9.8	10.2	11.3	12.4	13.5	14.6	16.8	19.0
	9	9.2	9.4	9.6	9.8	10.0	10.3	10.7	11.1	12.1	13.2	14.2	15.3	17.4	19.5
	10	10.2	10.4	10.6	10.8	11.0	11.2	11.6	12.0	13.0	14.0	15.0	16.0	18.0	20.0

Figures above heavy line are in pounds per square inch gauge.
Figures below heavy line are in inches of mercury vacuum.

TABLE XX

Capacity of 2–In. Pipe Line (2.067–In. Inside Diameter) with 0.75 Gravity Gas

Feet	Percentage of Pressure Drop in Pipe Line														
	0.5	1	2	3	4	5	6	8	10	15	20	25	30	40	50
100	3.90	5.50	7.76	9.48	10.9	12.2	13.3	15.3	17.0	20.6	23.4	25.8	27.9	31.2	33.8
200	2.76	3.88	5.48	6.70	7.72	8.61	9.40	10.8	12.0	14.5	16.5	18.2	19.7	22.1	23.9
300	2.25	3.18	4.47	5.48	6.31	7.03	7.69	8.84	9.83	11.9	13.5	14.9	16.1	18.0	19.5
400	1.95	2.75	3.88	4.74	5.46	6.08	6.65	7.64	8.50	10.3	11.7	12.9	13.9	15.6	16.9
500	1.75	2.46	3.47	4.23	4.88	5.43	5.95	6.83	7.60	9.19	10.5	11.5	12.5	14.0	15.1
600	1.59	2.24	3.16	3.87	4.45	4.97	5.43	6.23	6.94	8.38	9.55	10.5	11.4	12.7	13.8
700	1.47	2.08	2.97	3.58	4.03	4.60	5.03	5.78	6.43	7.77	8.85	9.75	10.5	11.8	12.8
800	1.38	1.95	2.75	3.35	3.87	4.31	4.71	5.41	6.02	7.28	8.28	9.13	9.86	11.0	12.0
900	1.29	1.83	2.58	3.16	3.64	4.06	4.43	5.09	5.67	6.85	7.80	8.59	9.28	10.4	11.2
1,000	1.23	1.74	2.45	2.99	3.45	3.85	4.20	4.83	5.38	6.49	7.39	8.15	8.80	9.86	10.7
1,200	1.13	1.59	2.24	2.74	3.16	3.52	3.85	4.42	4.92	5.93	6.76	7.45	8.05	9.02	9.76
1,400	1.04	1.47	2.07	2.53	2.92	3.25	3.54	4.08	4.54	5.48	6.25	6.88	7.43	8.33	9.02
1,600	0.98	1.37	1.94	2.37	2.73	3.04	3.33	3.82	4.25	5.13	5.85	6.44	6.96	7.80	8.44
1,800	0.92	1.29	1.83	2.24	2.58	2.88	3.14	3.61	4.02	4.85	5.52	6.08	6.58	7.37	7.98
2,000	0.87	1.23	1.74	2.12	2.45	2.72	2.98	3.43	3.81	4.60	5.25	5.78	6.24	6.99	7.57
2,200	0.83	1.17	1.65	2.02	2.31	2.59	2.84	3.26	3.63	4.38	4.98	5.48	5.93	6.64	7.19
2,400	0.80	1.11	1.58	1.93	2.22	2.48	2.71	3.12	3.47	4.19	4.77	5.27	5.68	6.36	6.89
2,600	0.76	1.07	1.52	1.86	2.14	2.38	2.61	2.99	3.33	4.02	4.58	5.05	5.45	6.11	6.62
2,800	0.73	1.04	1.47	1.79	2.06	2.30	2.52	2.89	3.22	3.89	4.42	4.87	5.27	5.90	6.38
3,000	0.71	1.00	1.42	1.73	2.00	2.22	2.43	2.80	3.11	3.76	4.28	4.72	5.09	5.71	6.18
3,500	0.66	0.93	1.31	1.60	1.84	2.06	2.24	2.58	2.88	3.47	3.96	4.36	4.70	5.27	5.70
4,000	0.62	0.87	1.23	1.50	1.72	1.92	2.10	2.41	2.69	3.25	3.70	4.06	4.40	4.93	5.34
4,500	0.58	0.82	1.16	1.41	1.63	1.81	1.98	2.28	2.54	3.06	3.49	3.84	4.15	4.65	5.03
5,000	0.55	0.78	1.09	1.34	1.54	1.72	1.87	2.15	2.40	2.90	3.30	3.64	3.93	4.40	4.76
6,000	0.51	0.71	1.00	1.22	1.41	1.56	1.71	1.97	2.19	2.65	3.02	3.32	3.59	4.03	4.36
7,000	0.47	0.66	0.93	1.14	1.31	1.46	1.59	1.84	2.04	2.47	2.81	3.10	3.35	3.75	4.05
8,000	0.44	0.62	0.87	1.06	1.22	1.36	1.49	1.71	1.91	2.30	2.62	2.89	3.12	3.50	3.78
9,000	0.41	0.58	0.82	0.99	1.14	1.27	1.39	1.61	1.78	2.15	2.46	2.71	2.92	3.28	3.55
10,000	0.39	0.55	0.78	0.95	1.09	1.22	1.33	1.53	1.70	2.06	2.34	2.58	2.78	3.12	3.37

Multiply tabular values by initial gauge pressure of line + 14.73 lb. per sq. in. to obtain capacity of line in thousands of cubic feet per 24 hours (M.c.f.).

TABLE XXI

Capacity of $2\frac{1}{2}$-In. Pipe Line (2.469-In. Inside Diameter) with 0.75 Gravity Gas

Feet	Percentage of Pressure Drop in Pipe Line														
	0.5	1	2	3	4	5	6	8	10	15	20	25	30	40	50
100	6.27	8.85	12.5	15.2	17.5	19.6	21.4	24.6	27.3	33.0	37.6	41.5	44.8	50.2	54.3
200	4.44	6.25	8.83	10.8	12.4	13.8	15.1	17.4	19.3	23.4	26.6	29.3	31.7	35.5	38.4
300	3.62	5.12	7.22	8.82	10.2	11.3	12.4	14.2	15.8	19.1	21.7	23.9	25.9	29.0	31.4
400	3.13	4.42	6.24	7.62	8.78	9.78	10.7	12.3	13.7	16.5	18.8	20.7	22.4	25.1	27.2
500	2.81	3.96	5.58	6.82	7.86	8.75	9.57	11.0	12.2	14.8	16.8	18.5	20.0	22.4	24.3
600	2.56	3.61	5.10	6.22	7.17	8.00	8.73	10.0	11.1	13.5	15.3	16.9	18.3	20.5	22.2
700	2.37	3.34	4.72	5.75	6.64	7.40	8.09	9.30	10.3	12.5	14.2	15.7	16.9	19.0	20.5
800	2.22	3.13	4.42	5.40	6.22	6.93	7.58	8.70	9.68	11.7	13.3	14.7	16.0	17.8	19.2
900	2.08	2.94	4.16	5.08	5.85	6.52	7.12	8.20	9.11	11.0	12.5	13.8	14.8	16.7	18.1
1,000	1.98	2.80	3.95	4.82	5.56	6.18	6.77	7.77	8.65	10.4	11.9	13.1	14.1	15.8	17.1
1,200	1.82	2.55	3.61	4.40	5.08	5.67	6.18	7.12	7.90	9.55	10.9	12.0	12.9	14.5	15.7
1,400	1.67	2.36	3.33	4.07	4.69	5.23	5.70	6.57	7.30	8.72	10.0	11.1	11.9	13.4	14.5
1,600	1.57	2.21	3.12	3.81	4.38	4.90	5.35	6.15	6.72	8.26	9.41	10.4	11.2	12.5	13.6
1,800	1.48	2.08	2.94	3.60	4.14	4.62	5.05	5.81	6.46	7.80	8.88	9.80	10.6	11.8	12.8
2,000	1.40	1.98	2.80	3.41	3.93	4.38	4.79	5.51	6.13	7.40	8.44	9.28	10.0	11.2	12.2
2,200	1.34	1.88	2.67	3.24	3.75	4.17	4.57	5.24	5.83	7.04	8.02	8.83	9.54	10.7	11.6
2,400	1.28	1.79	2.54	3.11	3.58	4.00	4.36	5.02	5.57	6.75	7.68	8.46	9.13	10.2	11.1
2,600	1.22	1.73	2.44	2.99	3.44	3.83	4.18	4.82	5.36	6.47	7.37	8.13	8.87	9.83	10.6
2,800	1.18	1.67	2.36	2.88	3.32	3.70	4.05	4.65	5.17	6.25	7.12	7.83	8.47	9.48	10.3
3,000	1.15	1.62	2.28	2.79	3.21	3.58	3.91	4.51	5.01	6.05	5.89	7.59	8.19	9.18	9.93
3,500	1.06	1.49	2.11	2.58	2.96	3.31	3.61	4.16	4.62	5.58	6.37	7.02	7.57	8.48	9.17
4,000	0.99	1.39	1.97	2.41	2.78	3.09	3.38	3.88	4.32	5.22	5.93	6.53	7.08	7.92	8.58
4,500	0.93	1.31	1.86	2.27	2.62	2.92	3.19	3.67	4.08	4.93	5.61	6.18	6.58	7.47	8.08
5,000	0.88	1.25	1.75	2.15	2.47	2.76	3.02	3.47	3.86	4.67	5.30	5.85	6.32	7.07	7.65
6,000	0.81	1.14	1.60	1.96	2.26	2.52	2.75	3.16	3.52	4.27	4.86	5.35	5.77	6.47	7.01
7,000	0.76	1.06	1.49	1.83	2.11	2.35	2.56	2.95	3.29	3.97	4.52	4.98	5.38	6.03	6.52
8,000	0.70	0.99	1.41	1.71	1.97	2.19	2.39	2.75	3.06	3.70	4.21	4.65	5.02	5.63	6.08
9,000	0.66	0.93	1.31	1.59	1.84	2.05	2.24	2.58	2.88	3.46	3.96	4.36	4.70	5.27	5.70
10,000	0.63	0.88	1.25	1.52	1.75	1.96	2.14	2.46	2.73	3.30	3.76	4.15	4.48	5.02	5.43
15,000	0.51	0.72	1.03	1.25	1.44	1.60	1.75	2.02	2.24	2.71	3.09	3.40	3.68	4.11	4.46
20,000	0.45	0.62	0.88	1.08	1.24	1.38	1.51	1.74	1.93	2.34	2.66	2.93	3.17	3.55	3.84
25,000	0.39	0.56	0.79	0.96	1.10	1.24	1.35	1.55	1.73	2.08	2.37	2.62	2.82	3.16	3.42
30,000	0.37	0.51	0.72	0.88	1.01	1.13	1.24	1.42	1.58	1.91	2.17	2.40	2.59	2.90	3.14

Multiply tabular values by initial gauge pressure of line + 14.73 lb. per sq. in. to obtain capacity of line in thousands of cubic feet per 24 hours (M.c.f.).

TABLE XXII

CAPACITY OF 3-IN. PIPE LINE (3.068-IN. INSIDE DIAMETER) WITH 0.75 GRAVITY GAS

Feet	Percentage of Pressure Drop in Pipe Line														
	0.5	1	2	3	4	5	6	8	10	15	20	25	30	40	50
100	11.2	15.8	22.2	27.2	31.3	34.9	38.1	43.8	48.8	58.9	67.1	74.0	79.9	89.5	96.9
200	7.91	11.1	15.7	19.2	22.1	24.7	27.0	31.0	34.5	41.7	47.4	52.3	56.5	63.3	68.5
300	6.46	9.12	12.8	15.7	18.1	20.2	22.1	25.3	28.2	34.1	38.8	42.7	46.2	51.7	56.0
400	5.58	7.90	11.1	13.6	15.7	17.5	19.1	21.9	24.4	29.5	33.6	37.0	39.9	44.7	48.4
500	5.01	7.05	9.95	12.2	14.0	15.6	17.0	19.6	21.8	26.3	30.0	33.1	35.7	40.0	43.3
600	4.57	6.44	9.08	11.1	12.8	14.2	15.6	17.9	19.9	24.0	27.4	30.2	32.6	36.5	39.5
700	4.23	5.96	8.40	10.3	11.8	13.2	14.4	16.6	18.3	22.3	25.4	28.0	30.2	33.8	36.6
800	3.95	5.58	7.90	9.62	11.1	12.3	13.5	15.5	17.3	20.9	23.7	26.2	28.3	31.7	34.3
900	3.72	5.25	7.42	9.06	10.4	11.6	12.7	14.6	16.2	19.6	22.3	24.6	26.6	29.8	32.3
1,000	3.54	4.99	7.03	8.58	9.89	11.0	12.1	13.8	15.4	18.6	21.2	23.4	25.2	28.3	30.7
1,200	3.24	4.55	6.45	7.85	9.06	10.1	11.0	12.7	14.1	17.0	19.4	21.4	23.1	25.9	28.0
1,400	2.98	4.22	5.95	7.25	8.36	9.34	10.2	11.7	13.0	15.7	17.9	19.7	21.3	23.9	25.9
1,600	2.81	3.94	5.57	6.80	7.83	8.73	9.55	11.0	12.2	14.7	16.8	18.5	19.9	22.3	24.2
1,800	2.64	3.72	5.25	6.42	7.40	8.25	9.01	10.3	11.5	13.9	15.8	17.4	18.9	21.1	22.9
2,000	2.50	3.54	4.98	6.08	7.02	7.82	8.55	9.82	10.9	13.2	15.0	16.5	17.9	20.0	21.7
2,200	2.38	3.36	4.73	5.78	6.68	7.43	8.13	9.35	10.4	12.5	14.3	15.7	17.0	19.0	20.6
2,400	2.28	3.20	4.53	5.55	6.38	7.12	7.77	8.95	9.94	12.0	13.7	15.1	16.3	18.2	19.7
2,600	2.18	3.08	4.35	5.32	6.14	6.83	7.47	8.58	9.56	11.5	13.1	14.5	15.6	17.5	19.0
2,800	2.10	2.98	4.22	5.13	5.93	6.60	7.22	8.30	9.23	11.1	12.7	14.0	15.1	16.9	18.3
3,000	2.05	2.88	4.07	4.97	5.73	6.38	6.97	8.03	8.93	10.8	12.3	13.5	14.6	16.4	17.7
3,500	1.89	2.66	3.76	4.59	5.29	5.90	6.44	7.42	8.25	9.95	11.3	12.5	13.5	15.1	16.3
4,000	1.77	2.48	3.52	4.30	4.95	5.50	6.02	6.92	7.72	9.31	10.6	11.6	12.6	14.1	15.3
4,500	1.67	2.34	3.32	4.05	4.67	5.21	5.68	6.54	7.27	8.78	10.0	11.0	11.9	13.3	14.4
5,000	1.57	2.22	3.12	3.84	4.41	4.93	5.39	6.18	6.87	8.31	9.46	10.4	11.3	12.6	13.6
6,000	1.45	2.03	2.86	3.50	4.03	4.49	4.91	5.65	6.28	7.62	8.66	9.55	10.3	11.5	12.5
7,000	1.35	1.89	2.66	3.26	3.76	4.20	4.57	5.27	5.87	7.08	8.05	8.90	9.62	10.7	11.6
8,000	1.25	1.77	2.50	3.04	3.52	3.92	4.28	4.92	5.47	6.60	7.52	8.29	8.95	10.0	10.8
9,000	1.17	1.65	2.34	2.84	3.28	3.66	4.00	4.62	5.13	6.18	7.06	7.78	8.40	9.41	10.2
10,000	1.12	1.58	2.22	2.72	3.13	3.49	3.81	4.38	4.88	5.89	6.71	7.40	7.99	8.95	9.69
15,000	0.91	1.29	1.83	2.22	2.56	2.86	3.12	3.60	4.00	4.83	5.61	6.06	6.56	7.33	7.95
20,000	0.79	1.11	1.57	1.92	2.21	2.47	2.70	3.10	3.45	4.17	4.75	5.23	5.65	6.33	6.85
25,000	0.70	0.99	1.41	1.71	1.96	2.21	2.41	2.76	3.08	3.72	4.23	4.67	5.03	5.65	6.10
30,000	0.65	0.91	1.28	1.57	1.81	2.02	2.21	2.53	2.82	3.41	3.88	4.27	4.62	5.17	5.60
50,000	0.50	0.70	0.99	1.22	1.40	1.56	1.70	1.96	2.18	2.63	3.00	3.31	3.57	4.00	4.33
100,000	0.35	0.50	0.70	0.86	0.99	1.10	1.21	1.38	1.54	1.86	2.12	2.34	2.52	2.83	3.07

Multiply tabular values by initial gauge pressure of line + 14.73 lb. per sq. in. to obtain capacity of line in thousands of cubic feet per 24 hours (M.c.f.).

TABLE XXIII

Capacity of 4–In. Pipe Line (4.026–In. Inside Diameter) with 0.75 Gravity Gas

Feet	Percentage of Pressure Drop in Pipe Line														
	0.5	1	2	3	4	5	6	8	10	15	20	25	30	40	50
100	23.1	32.6	45.9	56.1	64.6	72.0	78.8	90.5	100.7	121.7	138.6	152.7	164.9	184.7	200.0
200	16.3	23.0	32.5	39.7	45.7	50.9	55.7	64.0	71.2	86.0	98.0	107.9	116.6	130.6	141.3
300	13.3	18.8	26.5	32.4	37.4	41.6	45.5	52.3	58.2	70.3	80.1	88.2	95.3	106.8	115.6
400	11.5	16.3	23.0	28.1	32.3	36.0	39.4	45.2	50.3	60.8	69.3	76.3	82.4	92.4	100.0
500	10.3	14.6	20.5	25.1	28.9	32.2	35.2	40.5	45.0	54.4	61.9	68.3	73.7	82.6	89.4
600	9.43	13.3	18.7	22.9	26.4	29.4	32.1	36.9	41.1	49.6	56.5	62.3	67.3	75.3	81.6
700	8.75	12.3	17.3	21.2	24.4	27.2	29.8	34.2	38.1	46.0	52.4	57.7	62.3	69.8	75.6
800	8.15	11.5	16.3	19.8	22.9	25.5	27.9	32.0	35.6	43.1	49.0	54.0	58.4	65.4	70.8
900	7.67	10.8	15.3	18.7	21.5	24.0	26.2	30.1	33.5	40.5	46.1	50.8	54.9	61.5	66.6
1,000	7.30	10.3	14.5	17.7	20.4	22.8	24.9	28.6	31.8	38.4	43.8	48.2	52.1	58.4	63.2
1,200	6.68	9.40	13.3	16.2	18.7	20.8	22.8	26.2	29.1	35.1	40.0	44.1	47.7	53.4	57.8
1,400	6.15	8.70	12.3	15.0	17.3	19.2	21.0	24.2	26.9	32.5	37.0	40.7	44.0	49.3	53.4
1,600	5.78	8.11	11.5	14.0	16.2	18.0	19.7	22.6	25.1	30.4	34.6	38.1	41.2	46.1	50.0
1,800	5.45	7.67	10.8	13.3	15.2	17.0	18.6	21.4	23.7	28.7	32.7	36.0	38.9	43.6	47.2
2,000	5.17	7.30	10.3	12.5	14.5	16.1	17.6	20.3	22.6	27.2	31.0	34.2	36.9	41.4	44.8
2,200	4.92	6.93	9.76	11.9	13.8	15.3	16.8	19.3	21.4	25.9	29.5	32.5	35.1	39.3	42.6
2,400	4.72	6.60	9.35	11.4	13.2	14.7	16.0	18.5	20.5	24.8	28.3	31.1	33.6	37.7	40.8
2,600	4.51	6.35	9.00	11.0	12.7	14.1	15.4	17.7	19.7	23.8	27.1	29.9	32.3	36.2	39.2
2,800	4.35	6.15	8.70	10.6	12.2	13.6	14.9	17.1	19.0	23.0	26.2	28.8	31.2	34.9	37.8
3,000	4.22	5.95	8.42	10.2	11.8	13.2	14.4	16.6	18.4	22.3	25.3	27.9	30.1	33.8	36.6
3,500	3.90	5.50	7.75	9.48	10.9	12.2	13.3	15.3	17.0	20.5	23.4	25.8	27.8	31.2	33.8
4,000	3.65	5.12	7.25	8.85	10.2	11.3	12.4	14.3	15.9	19.2	21.9	24.0	26.1	29.2	31.6
4,500	3.45	4.83	6.85	8.37	9.64	10.7	11.7	13.5	15.0	18.1	20.6	22.7	24.6	27.5	29.8
5,000	3.24	4.60	6.44	7.92	9.10	10.2	11.1	12.8	14.2	17.1	19.5	21.5	23.3	26.0	28.2
6,000	3.00	4.18	5.90	7.22	8.33	9.27	10.1	11.6	13.0	15.7	17.9	19.7	21.2	23.8	25.8
7,000	2.79	3.90	5.50	6.73	7.75	8.65	9.44	10.9	12.1	14.6	16.6	18.3	19.8	22.2	24.0
8,000	2.58	3.65	5.17	6.28	7.25	8.08	8.82	10.1	11.3	13.6	15.5	17.1	18.5	20.7	22.4
9,000	2.42	3.40	4.84	5.85	6.76	7.55	8.28	9.50	10.6	12.8	14.6	16.0	17.3	19.4	21.0
10,000	2.31	3.26	4.59	5.61	6.46	7.20	7.88	9.05	10.1	12.2	13.8	15.3	16.5	18.5	20.0
15,000	1.88	2.66	3.77	4.59	5.29	5.90	6.45	7.42	8.25	9.97	11.4	12.5	13.5	15.1	16.4
20,000	1.63	2.30	3.25	3.97	4.57	5.09	5.57	6.40	7.12	8.65	9.85	10.8	11.7	13.1	14.2
25,000	1.43	2.05	2.91	3.53	4.06	4.55	4.96	5.70	6.35	7.67	8.73	9.64	10.4	11.6	12.6
30,000	1.33	1.88	2.65	3.24	3.74	4.16	4.55	5.23	5.82	7.03	8.01	8.82	9.53	10.7	11.6
50,000	1.03	1.46	2.05	2.51	2.89	3.22	3.52	4.05	4.50	5.44	6.19	6.83	7.37	8.26	8.94
100,000	0.73	1.03	1.45	1.77	2.04	2.28	2.49	2.86	3.18	3.84	4.38	4.82	5.21	5.84	6.32

Multiply tabular values by initial gauge pressure of line + 14.73 lb. per sq. in. to obtain capacity of line in thousands of cubic feet per 24 hours (M.c.f.).

TABLE XXIV

Capacity of 6-In. Pipe Line (6.065-In. Inside Diameter) with 0.75 Gravity Gas

Feet	Percentage of Pressure Drop in Pipe Line														
	0.5	1	2	3	4	5	6	8	10	15	20	25	30	40	50
100	68.9	97.1	137.0	167.3	192.8	214.8	234.9	270.0	300.3	362.9	413.2	455.3	491.8	551.0	596.3
200	48.7	68.6	96.9	118.3	136.3	151.9	166.0	190.8	212.2	256.5	292.1	321.8	347.7	389.5	421.6
300	39.7	56.1	79.1	96.7	111.4	124.2	135.8	156.0	173.6	209.8	238.8	263.1	284.2	318.4	344.7
400	34.4	48.5	68.5	83.6	96.4	107.4	117.4	134.9	150.1	181.4	206.6	227.6	245.9	275.5	298.2
500	30.8	43.4	61.3	74.9	86.2	96.0	105.0	120.7	134.2	162.2	184.7	203.6	219.8	246.4	266.7
600	28.1	39.6	55.8	68.2	78.5	87.6	95.8	110.1	122.4	148.0	168.6	185.7	200.6	224.7	243.3
700	26.0	36.7	51.7	63.2	72.8	81.1	88.7	101.9	113.5	137.1	156.2	172.1	185.8	208.2	225.4
800	24.3	34.3	48.5	59.2	68.2	76.0	83.1	95.5	106.3	128.4	146.2	161.1	174.1	195.0	211.1
900	22.9	32.3	45.6	55.7	64.2	71.5	78.2	89.8	99.9	120.9	137.6	151.6	163.8	183.5	198.7
1,000	21.8	30.7	43.3	52.8	60.9	67.9	74.2	85.3	94.9	114.6	130.5	143.9	155.3	174.1	188.4
1,200	19.9	27.9	39.6	48.3	55.7	62.2	67.8	78.0	86.7	104.8	119.3	131.5	142.1	159.3	172.4
1,400	18.4	25.9	35.6	44.7	51.5	57.4	62.6	72.1	80.2	96.9	110.4	121.5	131.3	147.0	159.3
1,600	17.2	24.2	34.2	41.8	48.2	53.7	58.7	67.5	75.0	90.7	103.3	113.7	122.9	137.5	149.0
1,800	16.3	22.9	32.3	39.5	45.5	50.8	55.5	63.7	70.8	85.6	97.5	107.3	116.0	130.0	140.8
2,000	15.4	21.8	30.7	37.4	43.2	48.1	52.6	60.5	67.3	81.2	92.6	102.0	110.0	123.5	133.5
2,200	14.7	20.7	29.1	35.6	41.2	45.8	50.1	57.5	64.0	77.3	88.0	96.9	104.7	117.3	127.0
2,400	14.1	19.7	27.9	34.1	39.3	43.8	47.8	55.1	61.2	74.0	84.3	92.8	100.3	112.3	121.5
2,600	13.5	19.0	26.8	32.8	37.8	42.1	46.0	52.8	58.8	71.1	81.0	89.2	96.2	107.8	116.8
2,800	13.0	18.3	25.9	31.8	36.4	40.7	44.4	51.0	56.8	68.6	78.2	86.0	93.0	104.0	112.6
3,000	12.6	17.7	25.1	30.6	35.2	39.3	43.0	49.5	55.0	66.5	75.7	83.3	90.0	100.8	109.0
3,500	11.6	16.4	23.1	28.3	32.5	36.3	39.6	45.7	50.8	61.2	69.8	77.0	84.3	93.1	100.7
4,000	10.9	15.3	21.6	26.4	30.5	33.9	37.1	42.6	47.5	57.3	65.2	71.7	77.7	87.0	94.2
4,500	10.3	14.4	20.4	25.0	28.5	32.0	35.0	40.2	44.7	54.1	61.5	67.8	73.3	82.0	88.8
5,000	9.6	13.7	19.2	23.6	27.2	30.3	33.2	38.1	42.3	51.2	58.2	64.2	69.4	77.7	84.0
6,000	8.9	12.5	17.6	21.5	24.8	27.6	30.2	34.7	38.6	46.8	53.3	58.7	63.4	71.1	77.0
7,000	8.3	11.6	16.4	20.1	23.1	25.8	28.1	32.4	36.1	43.5	49.5	54.7	59.1	66.2	71.5
8,000	7.8	10.9	15.4	18.7	21.7	24.1	26.3	30.2	33.6	40.6	46.3	51.0	55.0	61.7	66.8
9,000	7.2	10.2	14.4	17.5	20.2	22.5	24.6	28.4	31.6	38.1	43.5	47.8	51.6	57.8	62.6
10,000	6.8	9.7	13.7	16.7	19.3	21.5	23.5	27.0	30.0	36.3	41.3	45.5	49.2	55.1	59.6
15,000	5.6	8.0	11.2	13.7	15.8	17.6	19.2	22.1	24.6	29.7	33.9	37.5	40.3	45.2	48.9
20,000	4.9	6.9	9.7	11.8	13.6	15.2	16.6	19.1	21.2	25.6	29.2	32.2	34.8	38.9	42.2
25,000	4.3	6.1	8.7	10.5	12.1	13.6	14.8	17.0	19.0	22.9	26.1	28.8	30.9	34.7	37.5
30,000	4.0	5.6	7.9	9.7	11.1	12.4	13.6	15.6	17.4	21.0	23.9	26.3	28.4	31.8	34.5
50,000	3.1	4.3	6.1	7.5	8.6	9.6	10.5	12.1	13.4	16.2	18.5	20.4	22.0	24.6	26.7
100,000	2.2	3.1	4.3	5.3	6.1	6.8	7.4	8.5	9.5	11.5	13.0	14.4	15.5	17.4	18.8

Multiply tabular values by initial gauge pressure of line + 14.73 lb. per sq. in. to obtain capacity of line in thousands of cubic feet per 24 hours (M.c.f.).

TABLE XXV

Capacity of 8-In. Pipe Line (8.071-In. Inside Diameter) with 0.75 Gravity Gas

Feet	Percentage of Pressure Drop in Pipe Line														
	0.5	1	2	3	4	5	6	8	10	15	20	25	30	40	50
100	147	208	293	358	413	460	503	578	643	777	885	975	1054	1181	1278
200	104	147	207	253	292	325	355	408	454	549	625	689	744	834	903
300	85.2	120	169	207	238	266	290	334	371	449	511	563	608	682	738
400	73.6	104	146	179	206	230	251	289	321	388	442	487	526	590	639
500	66.0	93.0	131	160	184	205	224	258	287	347	395	436	471	527	571
600	60.3	84.8	119	146	168	187	205	235	262	317	361	397	429	481	521
700	55.9	78.7	111	135	156	174	190	218	243	293	334	368	398	446	483
800	52.2	73.7	105	127	146	163	178	204	227	275	313	345	373	417	452
900	49.0	69.2	97.8	119	137	153	167	192	214	258	294	324	350	393	425
1,000	46.6	65.8	92.8	113	130	145	159	182	203	245	279	308	332	373	403
1,200	42.8	60.0	85.0	103	119	133	145	167	185	224	255	281	304	341	369
1,400	39.3	55.6	78.3	95.7	110	123	134	154	171	207	236	260	281	315	341
1,600	37.0	51.9	73.4	89.7	103	115	125	144	160	194	221	243	263	294	319
1,800	34.9	49.0	69.2	84.7	97.5	108	118	136	151	183	208	230	248	278	301
2,000	33.0	46.7	65.8	80.2	92.6	104	112	129	144	174	198	218	235	264	286
2,200	31.5	44.3	62.4	76.3	88.0	98.0	107	123	137	165	188	207	224	251	272
2,400	30.1	42.2	59.8	73.2	84.1	93.9	102	118	131	158	180	198	214	240	260
2,600	28.8	40.6	57.4	70.3	81.0	90.2	98.5	113	126	152	173	191	206	231	250
2,800	27.8	39.3	55.6	67.6	78.1	87.0	95.2	109	121	147	167	184	199	223	241
3,000	27.0	38.0	53.7	65.5	75.5	84.1	92.0	106	117	142	162	178	192	216	233
3,500	24.9	35.1	49.6	60.5	69.7	77.8	85.0	97.7	108	131	149	164	178	199	215
4,000	23.3	32.8	46.4	56.7	65.3	72.6	79.5	91.2	101	122	139	153	166	186	201
4,500	22.0	30.9	43.8	53.4	61.5	68.7	75.0	86.2	96.0	116	132	145	157	175	190
5,000	20.7	29.4	41.2	50.6	58.2	65.0	71.0	81.5	90.7	109	124	137	148	166	180
6,000	19.1	26.7	37.7	46.2	53.2	59.3	64.7	74.5	82.9	100	114	125	135	152	165
7,000	17.8	24.9	35.1	43.0	49.6	55.2	60.3	69.5	77.3	93.3	106	117	126	141	153
8,000	16.5	23.3	33.0	40.1	46.4	51.7	56.3	64.7	72.0	87.0	99.2	109	117	132	143
9,000	15.4	21.7	30.9	37.5	43.3	48.3	52.7	60.8	67.7	81.5	92.0	102	110	124	134
10,000	14.8	20.8	29.3	35.9	41.3	46.0	50.3	57.8	64.3	77.8	88.5	97.5	105	118	127
15,000	12.1	17.0	24.1	29.4	33.8	37.8	41.2	47.5	52.7	63.7	72.7	80.0	86.5	97.6	104
20,000	10.4	14.7	20.8	25.3	29.2	32.5	35.6	40.9	45.4	55.0	62.6	69.0	74.5	83.4	90.3
25,000	9.2	13.1	18.6	22.5	25.7	29.1	31.7	36.4	40.6	49.0	55.8	61.6	66.3	74.5	80.4
30,000	8.5	12.0	17.0	20.7	23.9	26.6	29.1	33.4	37.2	44.9	51.1	56.4	60.9	68.2	73.9
50,000	6.6	9.3	13.1	16.0	18.5	20.6	22.5	25.9	28.7	34.7	39.6	43.6	47.1	52.8	57.1
100,000	4.7	6.6	9.3	11.3	13.0	14.5	15.9	18.3	20.3	24.5	28.0	30.8	33.3	37.3	40.4

Multiply tabular values by initial gauge pressure of line + 14.73 lb. per sq. in. to obtain capacity of line in thousands of cubic feet per 24 hours (M.c.f.).

TABLE XXVI

Capacity of 10-In. Pipe Line (10.192-In. Inside Diameter) with 0.75 Gravity Gas

Feet	Percentage of Pressure Drop in Pipe														
	0.5	1	2	3	4	5	6	8	10	15	20	25	30	40	50
100	275	388	547	668	770	857	938	1078	1199	1449	1650	1817	1963	2199	2380
200	195	274	387	472	545	608	664	763	847	1024	1166	1285	1388	1555	1682
300	159	224	316	387	445	497	543	623	694	837	953	1050	1134	1271	1376
400	137	194	274	334	385	429	469	539	600	725	825	909	981	1099	1190
500	123	173	245	299	345	384	420	482	537	648	739	814	877	983	1064
600	112	158	223	273	314	350	383	440	489	592	674	742	802	898	973
700	104	147	207	253	291	325	355	408	454	548	624	688	743	832	902
800	97	137	194	236	273	304	332	382	425	513	584	644	695	780	844
900	91	129	182	223	257	286	312	359	400	483	550	606	655	734	794
1,000	87	122	173	211	243	271	296	340	379	457	521	574	620	695	752
1,200	80	112	158	193	223	248	271	312	347	418	467	526	568	637	689
1,400	73	103	146	178	206	229	250	288	320	387	441	485	524	588	637
1,600	69	97	137	167	192	215	235	270	300	362	412	455	491	550	595
1,800	65	91	129	158	182	203	222	255	283	342	390	429	464	520	563
2,000	62	87	122	150	173	192	210	241	269	324	370	407	440	493	534
2,200	59	83	116	142	164	183	200	230	256	309	351	387	418	468	508
2,400	56	79	111	136	157	175	191	220	244	296	337	371	401	448	486
2,600	54	76	107	131	151	168	184	211	235	284	323	356	385	431	467
2,800	52	73	104	126	146	162	177	204	227	274	312	344	371	416	450
3,000	50	71	100	122	141	157	172	198	219	266	302	333	359	403	436
3,500	46	65	92	113	130	145	158	182	203	245	279	308	332	372	402
4,000	43	61	87	105	122	135	148	170	190	229	261	286	310	347	376
4,500	41	58	82	100	115	128	140	161	179	216	246	271	293	328	355
5,000	39	55	77	94	108	121	132	152	169	204	232	256	277	310	336
6,000	36	50	70	86	99	110	121	139	154	187	213	234	253	284	307
7,000	33	46	65	80	92	103	112	130	144	174	198	218	236	264	286
8,000	31	43	62	75	87	96	105	121	134	162	185	204	220	247	267
9,000	29	41	58	70	81	90	98	113	126	152	173	191	206	231	250
10,000	27	39	55	67	77	86	94	108	120	145	165	182	196	220	238
15,000	22	32	45	55	63	70	77	89	98	118	140	149	161	180	195
20,000	19	27	39	47	54	61	66	76	85	102	117	128	139	155	168
25,000	17	24	35	42	48	54	59	68	76	91	104	115	123	139	150
30,000	16	22	32	39	44	50	54	62	69	84	95	105	113	127	138
50,000	12	17	24	30	34	38	42	48	54	65	74	81	88	98	106
100,000	9	12	17	21	24	27	30	34	38	46	52	57	62	69	75

Multiply tabular values by initial gauge pressure of line + 14.73 lb. per sq. in. to obtain capacity of line in thousands of cubic feet per 24 hours (M.c.f.).

TABLE XXVII

Capacity of 12-In. Pipe Line (12.09-In. Inside Diameter) with 0.75 Gravity Gas

Feet	Percentage of Pressure Drop in Pipe														
	0.5	1	2	3	4	5	6	8	10	15	20	25	30	40	50
100	433	611	862	1053	1213	1352	1478	1699	1890	2284	2601	2866	3095	3468	3754
200	306	432	610	744	858	956	1045	1201	1336	1615	1839	2026	2188	2452	2653
300	250	353	497	609	702	781	855	982	1093	1320	1503	1656	1788	2004	2170
400	216	305	431	526	606	676	738	849	945	1142	1300	1433	1548	1734	1877
500	194	273	385	471	542	605	660	760	847	1021	1163	1281	1384	1551	1678
600	177	249	352	430	495	552	603	693	770	932	1061	1169	1263	1414	1531
700	164	231	326	398	458	511	558	642	715	863	983	1083	1170	1310	1419
800	153	216	305	373	430	478	523	601	669	809	920	1014	1095	1227	1329
900	144	203	287	351	404	450	492	566	628	760	866	953	1031	1155	1250
1,000	137	193	272	333	383	427	467	537	597	721	821	905	978	1096	1185
1,200	125	176	249	304	351	391	427	492	546	659	751	828	895	1002	1084
1,400	115	163	230	281	324	361	394	453	504	609	693	765	826	925	1002
1,600	108	152	215	263	303	338	369	425	472	570	650	715	773	866	938
1,800	102	144	203	249	286	319	349	401	446	538	613	676	730	818	886
2,000	97	137	193	235	272	302	331	380	423	511	582	642	693	777	840
2,200	92	130	183	224	259	288	315	362	402	486	553	610	658	738	800
2,400	89	124	175	215	247	275	301	346	385	465	530	584	631	707	765
2,600	85	119	168	206	238	265	289	332	370	447	510	561	605	678	735
2,800	82	115	163	198	229	256	279	321	357	432	491	542	585	655	708
3,000	79	111	157	192	222	247	270	311	346	418	476	523	565	635	686
3,500	73	103	145	177	205	229	249	287	319	385	439	484	522	586	633
4,000	68	96	136	166	192	213	233	268	299	361	410	451	488	547	593
4,500	65	91	128	157	181	201	220	253	282	340	387	427	461	516	558
5,000	61	87	121	148	171	191	208	239	266	322	366	404	437	488	528
6,000	56	78	111	135	156	174	190	219	243	295	335	369	399	447	484
7,000	52	73	103	126	145	162	177	204	227	274	312	344	372	416	450
8,000	48	68	97	118	136	151	165	190	211	255	291	321	346	389	420
9,000	45	64	91	110	127	141	155	178	198	239	273	301	325	364	394
10,000	43	61	86	105	121	135	148	170	189	228	260	287	309	347	375
15,000	35	50	71	86	99	111	121	139	155	187	213	235	254	284	308
20,000	31	43	61	74	86	95	104	120	134	161	184	202	219	245	265
25,000	27	38	55	66	76	86	93	107	119	144	164	181	195	218	236
30,000	25	35	50	61	70	78	85	98	109	132	150	165	179	200	217
50,000	19	27	38	47	54	60	66	76	85	102	116	128	138	155	168
100,000	14	19	27	33	38	43	47	54	60	72	82	90	98	109	118

Multiply tabular values by initial gauge pressure of line + 14.73 lb. per sq. in. to obtain capacity of line in thousands of cubic feet per 24 hours (M.c.f.).

TABLE XXVIII

Capacity of 16–In. Pipe Line (15.25–In. Inside Diameter) with 0.75 Gravity Gas

Feet	Percentage of Pressure Drop in Pipe Line														
	0.5	1	2	3	4	5	6	8	10	15	20	25	30	40	50
100	805	1135	1602	1956	2254	2511	2745	3157	3511	4243	4831	5323	5749	6441	6972
200	569	802	1133	1383	1593	1776	1941	2231	2481	2999	3415	3763	4064	4554	4928
300	465	656	925	1131	1303	1452	1587	1823	2029	2453	2792	3076	3322	3723	4030
400	402	568	801	978	1127	1256	1373	1577	1755	2121	2415	2661	2875	3221	3487
500	360	507	716	875	1008	1122	1227	1410	1568	1895	2160	2380	2570	2880	3115
600	329	463	653	798	919	1025	1120	1287	1431	1730	1971	2171	2345	2627	2845
700	304	429	605	740	852	950	1038	1193	1327	1603	1826	2012	2172	2434	2635
800	285	402	568	693	798	889	972	1117	1243	1502	1709	1883	2035	2280	2468
900	268	377	533	652	750	837	914	1051	1168	1413	1609	1772	1915	2145	2322
1,000	255	359	506	618	712	794	868	997	1110	1340	1526	1682	1816	2035	2202
1,200	233	327	463	565	652	727	794	912	1014	1225	1396	1537	1662	1862	2015
1,400	215	303	427	522	602	671	732	842	937	1133	1290	1420	1534	1719	1862
1,600	202	283	400	489	563	628	686	790	877	1060	1207	1330	1436	1609	1742
1,800	190	267	378	462	532	594	648	745	828	1001	1140	1256	1357	1520	1646
2,000	180	255	359	437	505	562	615	707	787	950	1083	1191	1287	1443	1562
2,200	171	242	340	417	480	535	585	672	748	905	1028	1133	1224	1372	1485
2,400	164	230	326	399	460	512	560	644	715	865	985	1086	1173	1314	1422
2,600	157	221	313	383	442	492	538	617	687	830	945	1044	1125	1262	1366
2,800	152	215	303	369	426	475	520	597	664	802	913	1007	1087	1217	1318
3,000	147	207	293	357	412	459	502	578	642	777	885	973	1050	1178	1276
3,500	136	193	270	330	380	425	463	533	593	716	816	900	971	1089	1178
4,000	127	179	253	309	356	396	433	498	555	670	763	838	908	1017	1100
4,500	120	169	239	292	336	375	409	470	523	632	720	793	857	960	1038
5,000	113	160	225	276	317	355	387	445	495	598	682	750	811	908	982
6,000	104	142	206	252	290	323	353	406	452	548	623	686	742	831	900
7,000	97	136	192	234	270	302	329	379	422	509	579	640	692	773	836
8,000	90	127	180	219	253	282	308	353	393	475	540	597	643	722	781
9,000	84	119	169	204	236	263	287	332	368	445	507	559	603	677	732
10,000	80	113	160	196	225	251	274	316	351	424	483	532	575	644	697
15,000	66	93	131	160	184	206	224	259	287	348	396	437	472	528	572
20,000	57	80	113	138	159	178	194	223	248	300	341	376	406	455	493
25,000	50	71	101	123	142	159	173	199	222	267	305	336	362	406	438
30,000	46	65	92	113	130	145	159	182	203	245	279	308	332	372	403
50,000	36	51	72	87	101	112	123	141	15[illegible]	189	216	238	257	288	311
100,000	25	36	51	62	71	79	87	100	111	134	153	168	182	203	220

Multiply tabular values by initial gauge pressure of line + 14.73 lb. per sq. in. to obtain capacity of line in thousands of cubic feet per 24 hours (M.c.f.).

TABLE XXIX

Capacity of 20–In. Pipe Line (19.182–In. Inside Diameter) with 0.75 Gravity Gas

Feet	Percentage of Pressure Drop in Pipe Line														
	0.5	1	2	3	4	5	6	8	10	15	20	25	30	40	50
100	1484	2093	2053	3607	4155	4630	5062	5819	6473	7823	8906	9813	10599	11875	12854
200	1049	1479	2088	2550	2937	3275	3578	4113	4575	5529	6296	6934	7493	8395	9086
300	856	1210	1706	2086	2402	2676	2927	3362	3741	4522	5147	5679	6125	6863	7430
400	740	1046	1476	1803	2078	2315	2531	2908	3235	3910	4453	4907	5300	5938	6428
500	664	935	1321	1614	1858	2070	2262	2602	2892	3496	3981	4387	4738	5310	5748
600	606	853	1205	1471	1695	1890	2064	2373	2639	3190	3633	4002	4324	4843	5244
700	562	791	1115	1363	1571	1751	1914	2199	2447	2956	3367	3710	4005	4488	4853
800	524	740	1046	1276	1471	1640	1793	2059	2291	2768	3151	3472	3752	4203	4551
900	493	696	983	1202	1384	1542	1685	1938	2154	2605	2966	3267	3530	3955	4282
1,000	469	662	933	1139	1313	1463	1600	1838	2046	2470	2813	3100	3348	3752	4060
1,200	430	603	854	1041	1202	1339	1463	1682	1869	2260	2573	2834	3064	3433	3715
1,400	395	558	787	962	1110	1236	1350	1553	1727	2088	2378	2618	2829	3169	3433
1,600	372	522	738	902	1039	1157	1266	1455	1616	1954	2225	2452	2647	2966	3211
1,800	350	492	696	852	981	1094	1194	1374	1527	1846	2101	2315	2502	2803	3035
2,000	332	469	662	807	930	1036	1134	1302	1450	1751	1996	2196	2373	2660	2879
2,200	316	446	627	767	885	986	1078	1239	1379	1666	1895	2088	2257	2528	2737
2,400	303	424	601	735	846	943	1031	1186	1318	1595	1817	2001	2162	2420	2620
2,600	290	408	577	706	815	906	991	1139	1268	1532	1745	1922	2075	2326	2518
2,800	279	395	558	680	785	875	957	1099	1223	1479	1682	1853	2004	2244	2428
3,000	271	382	540	659	758	846	925	1065	1184	1432	1629	1795	1938	2172	2352
3,500	250	353	498	608	700	782	854	982	1094	1321	1505	1658	1790	2006	2170
4,000	235	329	467	569	657	730	798	917	1023	1234	1405	1545	1674	1875	2030
4,500	221	311	440	537	619	690	754	867	965	1165	1326	1461	1579	1769	1914
5,000	208	295	413	508	585	653	714	820	912	1102	1255	1384	1495	1674	1811
6,000	192	269	379	463	535	595	651	748	833	1010	1150	1266	1366	1532	1658
7,000	179	250	353	432	498	556	606	698	777	938	1068	1178	1273	1426	1542
8,000	166	235	332	403	466	519	567	651	725	875	997	1099	1186	1331	1440
9,000	155	219	311	377	435	485	530	612	680	820	935	1031	1113	1247	1350
10,000	148	209	295	361	415	463	506	582	647	782	891	981	1060	1186	1284
15,000	121	171	242	295	340	380	413	477	530	640	730	804	870	973	1055
20,000	105	148	209	255	294	327	358	411	458	553	630	693	749	844	912
25,000	92	132	187	227	261	292	319	366	408	492	562	618	667	749	809
30,000	86	121	171	208	240	268	293	336	374	452	515	542	612	686	742
50,000	66	93	132	161	186	207	226	260	289	350	398	439	474	531	578
100,000	47	66	93	114	131	146	160	184	205	247	281	310	335	375	406

Multiply tabular values by initial gauge pressure of line + 14.73 lb. per sq. in. to obtain capacity of line in thousands of cubic feet per 24 hours (M.c.f.).

LOGARITHMS

To find the logarithm of 326.78:

In the table, opposite 32, in column 6, find	5132
In same row, proportional part 7, find	9
In same row, 1/10 of proportional part 8	1.1
Total, prefixing a decimal point,	.5142

This is the *mantissa*, or decimal part of the logarithm. It is *always positive.*

The *characteristic*, or whole number part of the logarithm, is—

1. *Positive for numbers greater than* 1. It is one less than the number of figures preceding the decimal point, e.g., 2 for 326.78 and 0 for 1.276.

2. *Negative for numbers less than* 1. It is (numerically) one more than the number of zeros immediately following the decimal point; thus $\bar{3}$ (namely -3) for 0.0027 and $\bar{1}$ for 0.27.

Thus the complete logarithm for 326.78 is 2.5142, and that for 0.0326 is $\bar{2}.5132$. This really means $-2 + 0.5132$, namely $-(1.4868)$.

To find the number corresponding to the logarithm 1.4129:

First consider the mantissa, 4129. The next lower mantissa in the table is that of 258. It is 4116, which needs 13 units more. In the same row, in the table of proportional parts, we find 12, in column 7, hence the fourth figure of the required number is 7. A 6, *in the next place* would be $1/10 \times 10 = 1$ unit more, making 13 units in all. The number sought is therefore 25.876, there being two figures preceding the decimal point, since the characteristic of the original logarithm is 1.

To multiply, add logarithms; to divide, subtract logarithms.

To extract square root, divide the complete logarithm of the number by 2. To extract cube root, divide the complete logarithm by 3. Check the final root, by a rough calculation. In extracting roots make the logarithm *all positive* or *all negative;* namely $\bar{2}.5132$ is written $-(1.4868)$. To extract the cube root we then divide by 3, thus obtaining $-(0.4956) = \bar{1}.5044$.

The *cologarithm* of a number is found by subtracting each figure of the logarithm from 9, except the last figure, which is subtracted from 10; then append -10.

Number	*Logarithm*	*Cologarithm*
326.78	2.5142	$7.4858 - 10 = \bar{3}.4858$

In a series of consecutive multiplications and divisions, use cologarithms for the divisions. *Add the cologarithm of each divisor instead of subtracting its logarithm.* Thus find $578 \times \frac{273}{298} \times \frac{746}{760}$:

	log 578 = 2.7619
	log 273 = 2.4362
	log 746 = 2.8727
log 298 = 2.4742	colog 298 = 7.5258 − 10
log 760 = 2.8808	colog 760 = 7.1192 − 10
	22.7158 − 20 = 2.7158

The result sought is 519.7

Natural Numbers.	0	1	2	3	4	5	6	7	8	9	Proportional Parts. 1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	4	8	12	17	21	25	29	33	37
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	4	8	11	15	19	23	26	30	34
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	3	7	10	14	17	21	24	28	31
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	3	6	10	13	16	19	23	26	29
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	3	6	9	12	15	18	21	24	27
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	3	6	8	11	14	17	20	22	25
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	3	5	8	11	13	16	18	21	24
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	2	5	7	10	12	15	17	20	22
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	2	5	7	9	12	14	16	19	21
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	2	4	7	9	11	13	16	18	20
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	10
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	7
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	7

Natural Numbers.	0	1	2	3	4	5	6	7	8	9	Proportional Parts.								
											1	2	3	4	5	6	7	8	9
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	5	6	7
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4	4	5	6	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3	4	4	5	5
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3	4	4	5	5
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2	3	3	4	4
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	3	4

INDEX